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SOLID LUBRICATION DESIGN METHODOLOGY

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16. Abstract
A single element traction rig was used to measure the traction forces at the contact of a ball against a flat disc at room temperature under combined rolling and sliding.. The load and speed conditions were selected to match those anticipated for bearing applications in adiabatic diesel engines. The test program showed that the magnitude of traction forces were almost the same for all the lubricants tested; a lubricant should, therefore, be selected on the basis of its ability to prevent wear of the contact surfaces. Traction vs. slide/roll ratio curves were similar to those for liquid lubricants but the traction forces were an order of magnitude higher. The test data was used to derive equations to predict traction force as a function of contact stress and rolling speed. Qualitative design guidelines for solid lubricated concentrated contacts are proposed.

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SOLID LUBRICATION DESIGN METHODOLOGY

1.0 INTRODUCTION

The U.S. Department of Energy (DOE) has initiated a heavy duty transport technology project directed towards providing a technology base for use in developing advanced heavy duty transport engines. The transportation sector currently uses nearly 27% of the total energy and 61% of all the petroleum consumed in the United States. While there have been significant improvements in the heavy duty fleets and the automotive segments of the transportation industry, the fleet fuel mileage has begun to level off. The improvements have resulted primarily because of a reduction in size and weight of automobiles. Future reductions in fuel mileage are expected to be realized primarily through advanced heavy duty engines. The heavy duty transport technology project focuses on developing Adiabatic Diesel Engine technology for long-haul trucks. An Adiabatic Diesel Engine -- defined as a highly insulated, uncooled diesel engine and its waste heat utilization system has -- potential for vastly lower fuel consumption because of improved engine efficiency through higher operating temperatures, reduced heat losses, lighter weight and higher exhaust energy recovery. The higher operating temperatures, however, pose difficult structural, material and lubrication problems.

Adiabatic diesel engines will require bearings and other load carrying contacts to operate over a temperature range that exceeds the capability of currently used liquid lubricants. Solid lubricants and high temperature bearing materials will be required for some applications. The Solid Lubrication Design Methodology program sponsored by DOE and monitored by NASA-Lewis Research Center is a two-phase program at SKF Industries, Inc. to evaluate solid lubricants and bearing material combinations under controlled test conditions representative of projected bearing requirements in adiabatic diesel engines and to develop qualitative and quantitative guidelines for the design of solid lubricated bearings, gears and other load carrying contacts. In Phase I, the candidate lubricant and material combinations were evaluated at room temperature to select the most promising combinations which will be tested in the 400°F to 1000°F temperature range during Phase II. The analytical models and design guidelines developed in Phase I will be refined in Phase II to include the effect of temperature.

This report describes the work done in Phase I of the program. Literature dealing with solid lubrication is reviewed in Section 2.0. The test apparatus and test procedure are described in Section 3.0. Results of a test program to screen

candidate lubricant-bearing material combinations are given in Section 4.0. The traction force measurements for the most promising lubricant-material combinations are described in Section 5.0. Analytical models based on the test data are described in Section 6.0 along with some general comments. The work to be done in Phase II of the program, currently in progress, is outlined in Section 7.0.

2.0 LITERATURE REVIEW

An extensive literature search was undertaken to evaluate the solid lubrication work done to date and to select ideas applicable to the design of solid lubrication systems for adiabatic diesel engines. A number of the National Technical Information Service (NTIS) published searches [1-9] were reviewed and potentially useful references were selected. Solid lubrication literature in the SKF Library was also reviewed. The results of the literature review are summarized below.

Solid lubricated bearing systems have been used in applications where extreme temperatures, either cryogenic or high temperatures, or other environmental conditions such as a vacuum rule out the use of fluid lubricants. The applications range from sliding bearings in spacecraft to high temperature high speed rolling element bearings in gas turbine engines. High temperature solid lubricants include Molybdenum Disulphide (MoS_2), Tungsten Disulphide (WS_2), Graphites, Graphie Flouride $(CF_x)_n$, Polyimides, Polyimide composites, soft oxides such as Lead Monoxide and stable flourides of some metals such as Lithium, Calcium and Barium [10-16]. The most frequent use of solid lubricants has been as coatings in sliding bearing applications. The use of solid lubricants in rolling element bearings has not been very successful. The bearings have been lubricated by cages made of self-lubricating materials [17-25] or by gas entrained

powder lubricants [26]. The primary problems in cage lubricated bearings have been cage instability, excessive cage wear, and loss of bearing clearances because of thermal instability and excessive build up of solid lubricant film on the raceways. Christy [21] observed that the life of a solid lubricated rolling element bearing depends on the formation of an initial run-in film and that an increase in bearing clearance is required to allow for thicker lubricant films. Stevens and Todd [20] reported a threshold load below which there is zero wear of steel races for bearings lubricated with composite lubricant cages and that the threshold value is lower than the load capacity of similar oil lubricated bearings. Below the threshold load, excessive build up of film eventually caused an increase in torque. Above the threshold level, the torque level remained the same but noise increased due to wear debris build up. They observed that the cage wear was proportional to the rate of depletion of the solid lubricant film on the ball and that the bearing torque due to traction forces is independent of speed. Todd [19] found that ball/raceway microslip left distinct wear marks in the ball track and that thin Polytetrafluoroethylene (PTFE) films have a transition maximum Hertz stress of about 1200 MPa below which they protect against wear. Above the transition stress, the substrate wear was a function of the contact stress.

He concluded that a lubricant transfer film should be less than 1 μm to ensure smooth bearing operation.

Review of the literature dealing with solid lubricated bearings indicated a lack of understanding of the behavior of solid lubricated rolling/sliding contacts. Most of the solid lubricated rolling element bearings to date are minor modifications of bearings designed for oil lubrication. There are no design criteria for optimizing the performance of solid lubricated rolling element contacts. Basic solid lubrication studies have concentrated primarily on sliding contacts using pin on disc experiments or some variations of sliding wear testers such as LFW-1 and Falex testers [27-32]. Sauer et al [28] found that coefficient of friction and the wear rate of solid lubricant films increase with film thickness. There is an optimum value of film thickness for maximum wear life. Sutor [29] developed wear equations for graphite fiber reinforced polyimide composites described in Reference [33] using test data from a dual rub shoe rotating disc machine but found that the equations were not applicable to conditions other than those used in the test matrix. Fusaro [30,31] showed that polyimides have a transition temperature above which the lubricants form a thin surface film which flows plastically and adheres to the substrate. Below the

transition temperature, the polyimides wear in a brittle manner, producing clumps of powdery wear particles which do not adhere well to the substrate. The clumps slide through the contact as clumps, causing wear at the interface. Sliney [32] studied the flow of solid lubricant particles through the contact between a pin sliding on a glass disc lubricated by powdered lubricant sprinkled ahead of the contact. He found that the particles are readily drawn into the contact and deformed into a thin plastic film. The lubricant in the contact flows like a paste. Graphite and MoS_2 powders behaved the same way, except that MoS_2 adhered to the surfaces better than the graphite did.

Dry lubricated rolling contacts have been studied using four ball testers. Yonushonis [34] used silicon nitride balls and cups lubricated by graphite cages to run tests at 2760 MPa (400 ksi) the maximum Hertz contact stress, at a rotational speed of 10,000 rpm and a temperature of 1000°F; P2003 graphite was found to be the most effective lubricant. Scott [35] studied the effectiveness of powdered lubricants and cage materials using a four ball tester. He found that the life of rolling contacts was limited by progressive surface roughening and wear which resulted in excessive noise, vibration and rough running; MoS_2 was found to be the most effective lubricant for reducing the wear.

While four ball testers are a useful tool for empirical screening of lubricants under conditions representative of rolling element bearings, they do not allow measurement of the variables needed for understanding the behavior of solid lubricated rolling element contacts. Measurement of the rolling element-raceway traction forces and the rolling element-cage pocket forces is required along with an understanding of lubricant film transfer from the cage pocket to the rolling element and the raceways. Single element test rigs allow a study of the required variables. A concentrated contact traction measurement apparatus developed at Georgia Institute of Technology [37] was used to measure the traction forces transmitted by a crowned roller/flat-disc contact lubricated by burnished and sputtered lubricant films. The tests were conducted at contact stresses of 0.79 GPa (115 ksi) and 1.07 GPa (155 ksi) with an average rolling velocity of 2 m/s (80 in/s). The lubricants tested were MoS_2 and a Westinghouse Compact (WSe_2/In/Ga). Traction measurements for sputtered films showed a limiting traction coefficient of 0.3 for MoS_2 and 0.2 for the Westinghouse Compact when the sliding velocity is greater than four percent of the rolling velocity. The limiting traction coefficient is defined as the ratio of the traction force to the normal load when the traction force levels off in a traction curve. The sputtered films on the

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roller debonded immediately on the start of tests, while the disc coatings remained in place without damage as long as the sliding velocity was less than three percent of rolling velocity. Burnished films gave slightly lower traction forces. Unlubricated 52100 steel elements gave a limiting traction coefficient of 0.4 at a contact pressure of 1.07 GPa. The traction forces were found to increase in the first 2000 revolutions of operation and then level off.

Kannel and Barber [36] used a ball bearing simulator to study the rolling element-cage pocket forces in a bearing with a self-lubricating cage and solid film lubricants on raceways. Ball-raceway traction coefficient for uncoated races lubricated by a cage made of a composite material [33] rubbing against the ball increased linearly with sliding up to 3% slide-to-roll ratio and then leveled off at a value of 0.5. The ball-cage pocket friction coefficient was approximately 0.06. Experiments with raceways coated with MoS_2 and the composite cage gave a ball-raceway traction coefficient of 0.18 until approximately 7% slide-to-roll ratio. The MoS_2 film apparently broke down at this point and resulted in a high traction coefficient; subsequent behavior of the contact was as for uncoated raceways.

Few theoretical analyses are available to aid in the design of solid lubricated rolling element bearings. Finkin [38] ana-

lyzed the influence of certain design and material parameters on the wear of ball bearings lubricated by solid lubricants coated on raceways and concluded that the current practice of converting conventional designs to use solid lubricants is not a good approach. Implications of his analysis for bearing design were to use a small number of larger balls, and closely conforming races and balls. The lubricant should have high ductility and fracture toughness to prevent debonding of lubricant film and adhesion high enough to keep the lubricant from being squeezed out of the contact but low enough to prevent significant sliding wear. Use of thin (less than 0.5 μm) films of precious metals and precious metal alloys to enhance lubricant transfer between cage and ball surfaces and between ball and races has also been recommended [39].

The literature review showed that very little data or design criteria were available to aid in the design of solid lubricated rolling/sliding contacts. Most of the available data was not obtained under conditions representative of rolling element bearing applications. That may explain the poor correlation between wear test machine data and actual bearing hardware. Tests with assembled and lubricated bearings are not useful for basic studies because of the difficulty of separating what is

being tested. Therefore, it was decided to use a single element test rig to test solid lubricants under load, speed and temperatures representative of bearing applications in adiabatic diesel engines. An existing high speed elastohydrodynamic (EHD) test rig built by SKF was modified to allow traction force measurement for solid lubricated contacts. The data would be used to develop analytical models to predict traction forces as a function of load, speed and temperature and to develop qualitative guidelines for designing solid lubricated rolling bearings.

3.0 TEST APPARATUS AND TEST PROCEDURE

3.10 Test Apparatus

A schematic of the test rig is shown in Figure 1. The test rig was designed to measure film thickness and traction force in an oil lubricated contact between a rotating disc and a ball or a roller under controlled amounts of rolling, sliding, spin, lateral slip and contact load. The disc and the rolling element specimens are driven by independent drives. The speed and load capabilities of the test rig are summarized in Table 1.

The disc spindle is driven by a 3 HP 3600 RPM variable speed DC motor. By using step-up pulleys, the disc can be driven up to 15,000 RPM. The DC motor speed is controlled through a regenerative controller to allow traction measurements with the disc surface speed slower than the rolling element surface speed. The rolling element is mounted directly on a high speed grinding spindle capable of operating up to 42,000 RPM. The high frequency spindle motor is housed such that the axis of rotation of the rolling element is horizontal and passes through the center of the disc. The complete motor assembly can be moved horizontally in its housing to change the radius of the contact circle on the disc. The motor assembly and its housing are mounted on a vertical shaft supported by hydrostatic bearings to minimize

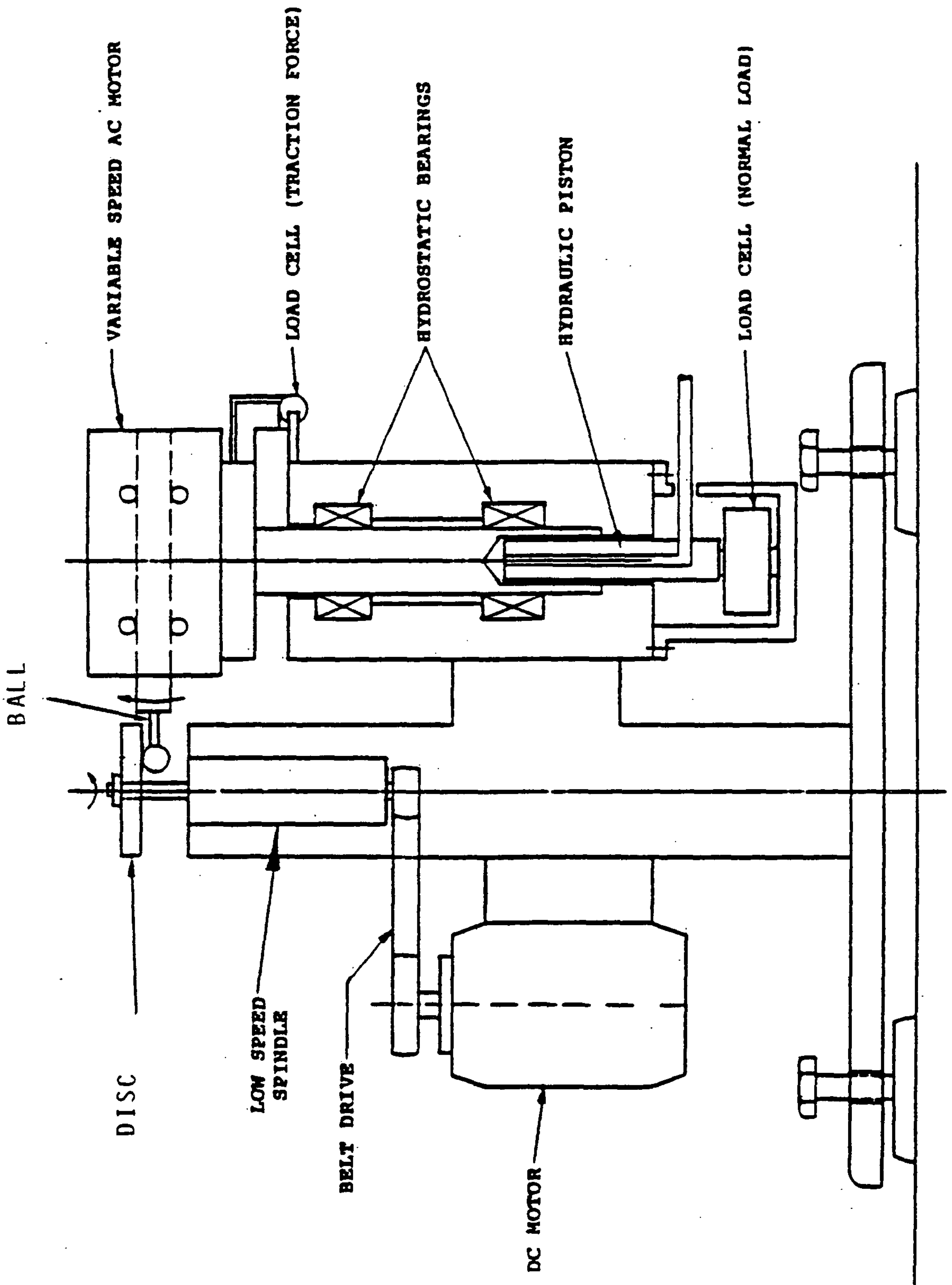

FIGURE 1: SCHEMATIC OF TEST RIG

TABLE 1: RIG CAPABILITIES

DISC DRIVE

VARIABLE SPEED REGENERATIVE DC MOTOR (3 HP, 3600 RPM)
STEP UP PULLEYS: DISC SPEED UP TO 15,000 RPM
SPEED CONTROL: ± 0.5%

BALL SPINDLE

VARIABLE SPEED AC MOTOR: 5.9 HP, 42,000 RPM
SPEED CONTROL: ± 0.1%

TRACTION LOAD CELL

DATA INSTRUMENTS LOAD CELL - MODEL NO. JP-50
50 LB. CAPACITY, ± 0.15%

NORMAL LOAD CELL

LEBOW MODEL NO. 3397 (STRAIN GAGE LOAD CELL)
300 LB. CAPACITY, ± 0.2%
VISHAY INSTRUMENTS V/E-20A STRAIN GAGE INDICATOR

SLIDE/ROLL RATIO: 0.1 MAX
LATERAL SLIP/ROLL RATIO: 0.02 MAX

the resistance to rotation of the rolling element about the vertical axis of the shaft. When a traction force acts on the rolling element, it attempts to rotate the motor assembly about the vertical axis of the shaft but is prevented from doing so by a semi-conductor load cell. The reactive force measured by the load cell is used to obtain the traction force in the contact. The normal load is applied by a hydraulic piston rod through the vertical shaft supporting the rolling element motor assembly. A strain gage load cell is mounted below the hydraulic piston to measure the normal load.

The EHD rig was modified to allow its use with solid lubricants. The lubricating oil supply and recirculation system was disconnected. Seals shown in Figure 2 were installed on the disc and rolling element spindles to keep the motor bearing lubricants from contaminating the contact.

A lubricant application fixture, shown in Figure 3, was designed to burnish the lubricant on the disc surface before the surface entered the contact. The relative positions of the contact and the lubricant burnishing location are shown in Figure 4. The lubricant is loaded against the disc to burnish the lubricant on the disc surface; the burnishing load can be changed by changing the weight on the lever arm (see Figure 3). A photograph of the modified test rig is shown in Figure 5.

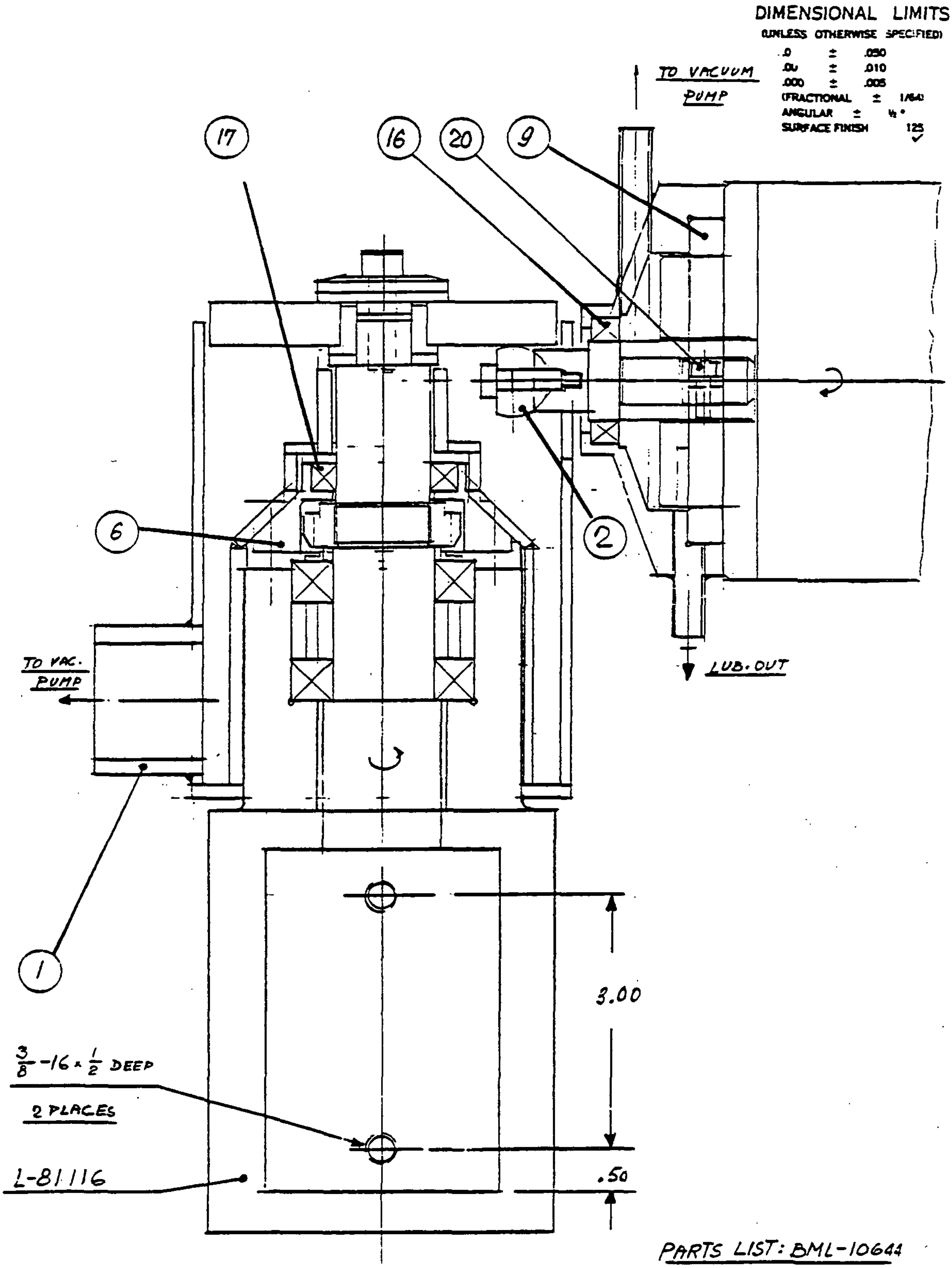

FIGURE 2: DRAWING OF LOW SPEED, 9 , AND HIGH SPEED, 17 , SEALS

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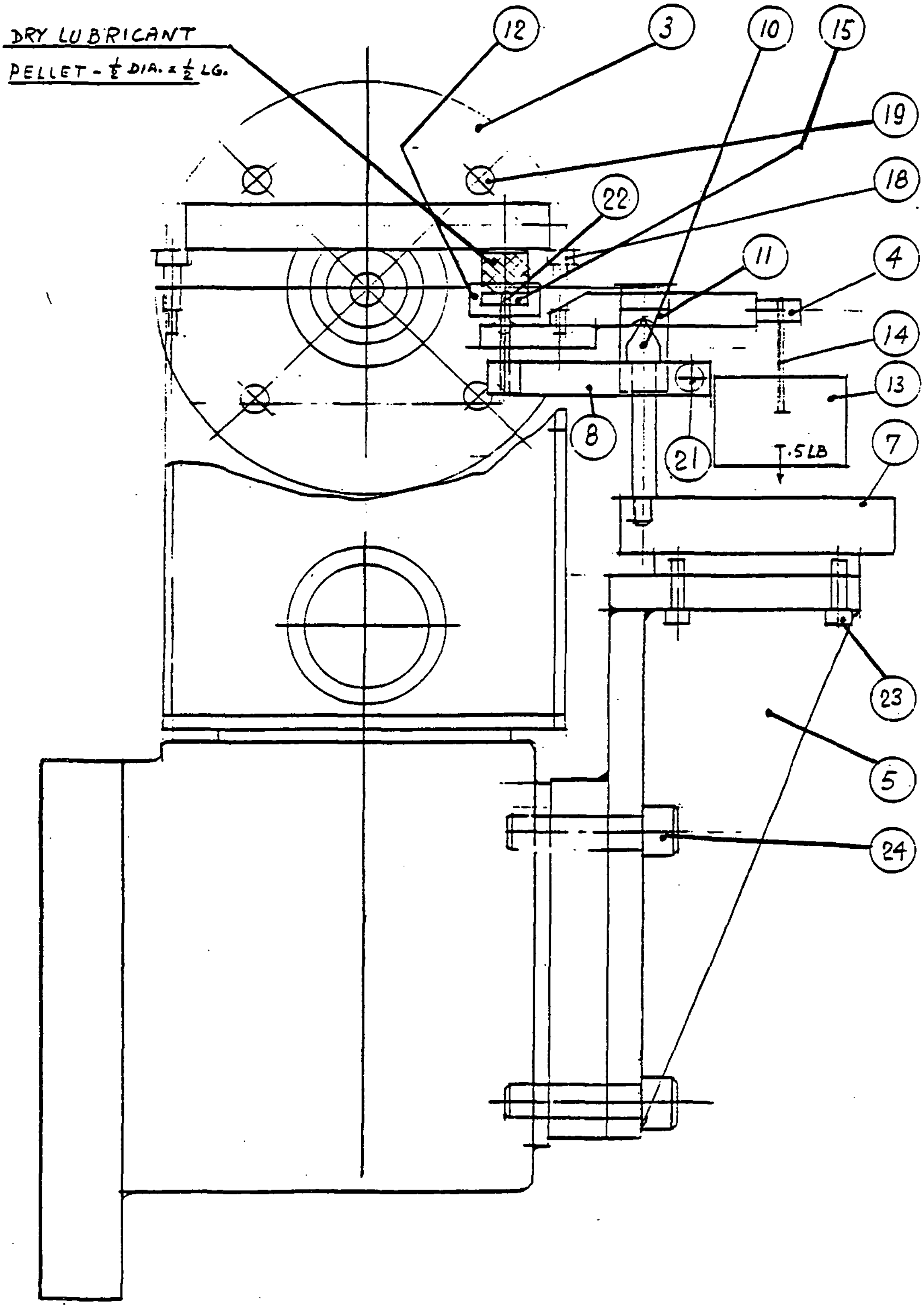
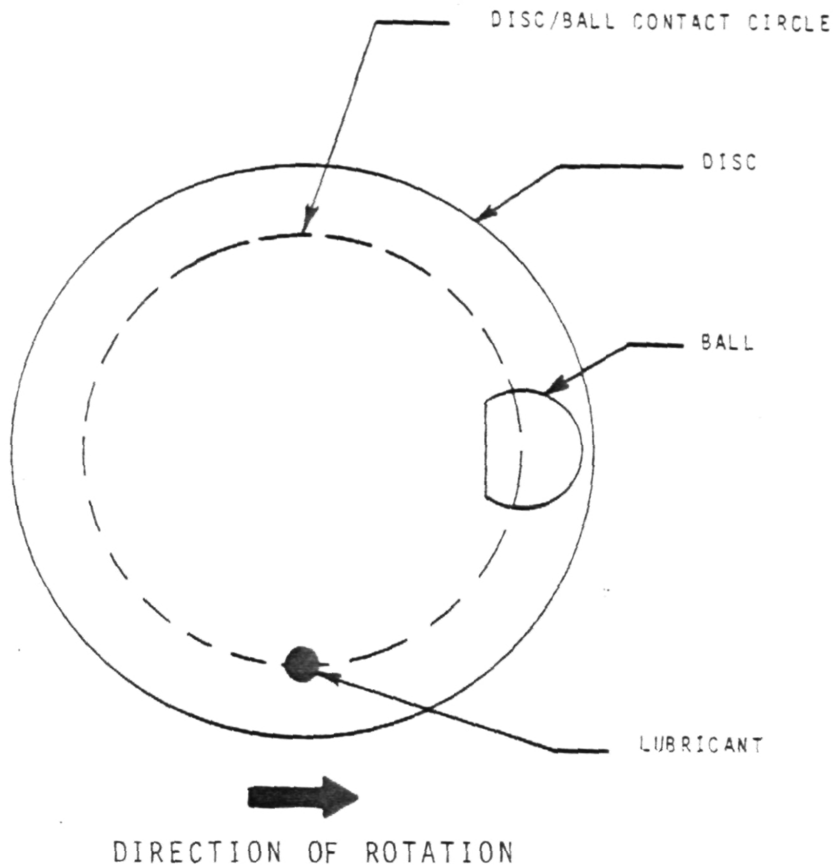


FIGURE 3: DRAWING OF LUBRICANT TRANSFER SYSTEM



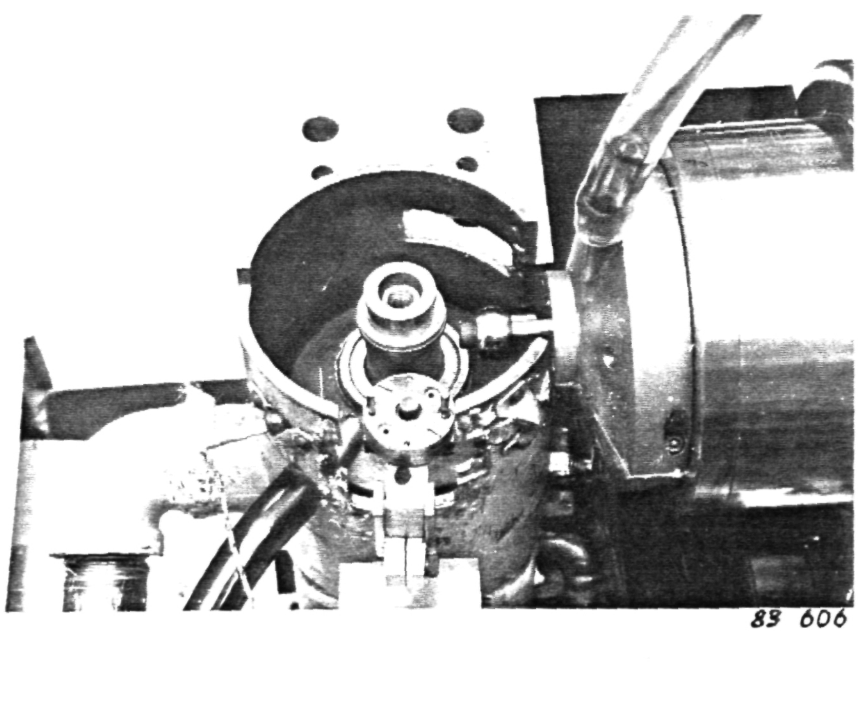


FIGURE 4: LUBRICANT BURNISHING ARRANGEMENT

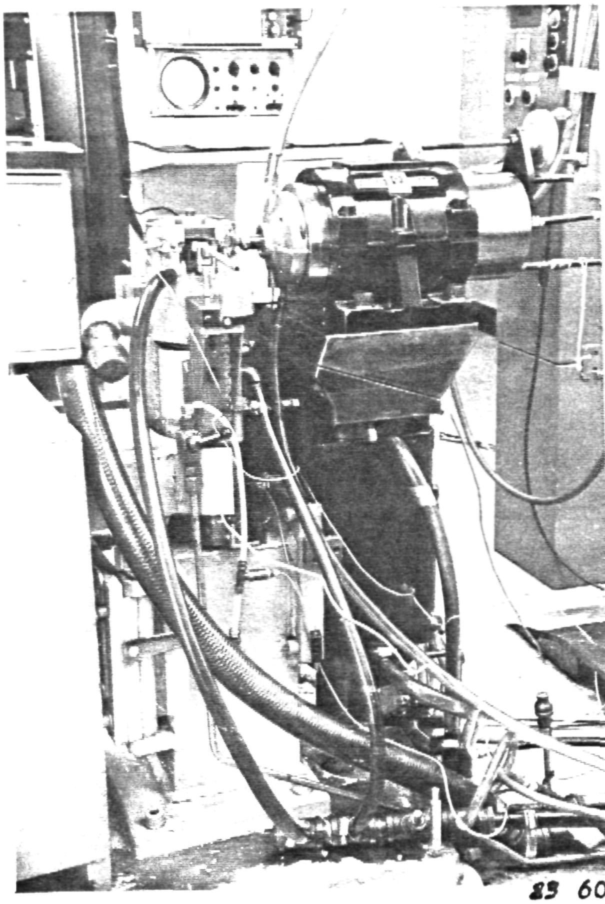


FIGURE 5: TEST RIG AS MODIFIED FOR SOLID LUBRICANT TESTING

3.11 Traction Measurements

Contact geometry is shown in Figure 6. Only ball specimens were used in solid lubricated traction measurements. The slide/roll ratio S at the contact is given by

$$S = \frac{2[\omega_1 - (D/d)\omega_2]}{\omega_1 + (D/d)\omega_2} \tag{1}$$

where ω_1 is the speed of rotation of the disc, ω_2 the speed of rotation of the ball, D the diameter of the ball and d the diameter of the contact circle on the disc (see Figure 4). The slide/roll ratio is positive when the disc surface is faster than the ball surface and negative when the disc surface is slower than the ball surface.

The disc spindle has a 6 tooth pulley for speed measurement. A proximity probe is placed near the pulley to generate a digital pulse train. The high speed spindle has direct digital speed output. The two outputs are fed into an electronic circuit of SKF design to give directly the slide/roll ratio defined in Equation (1). This signal drives the x-axis of an x-y plotter. The individual speeds of the drives are read from electronic counters. The output from the traction load cell is fed to the y axis of the x-y plotter. Thus, the traction force versus

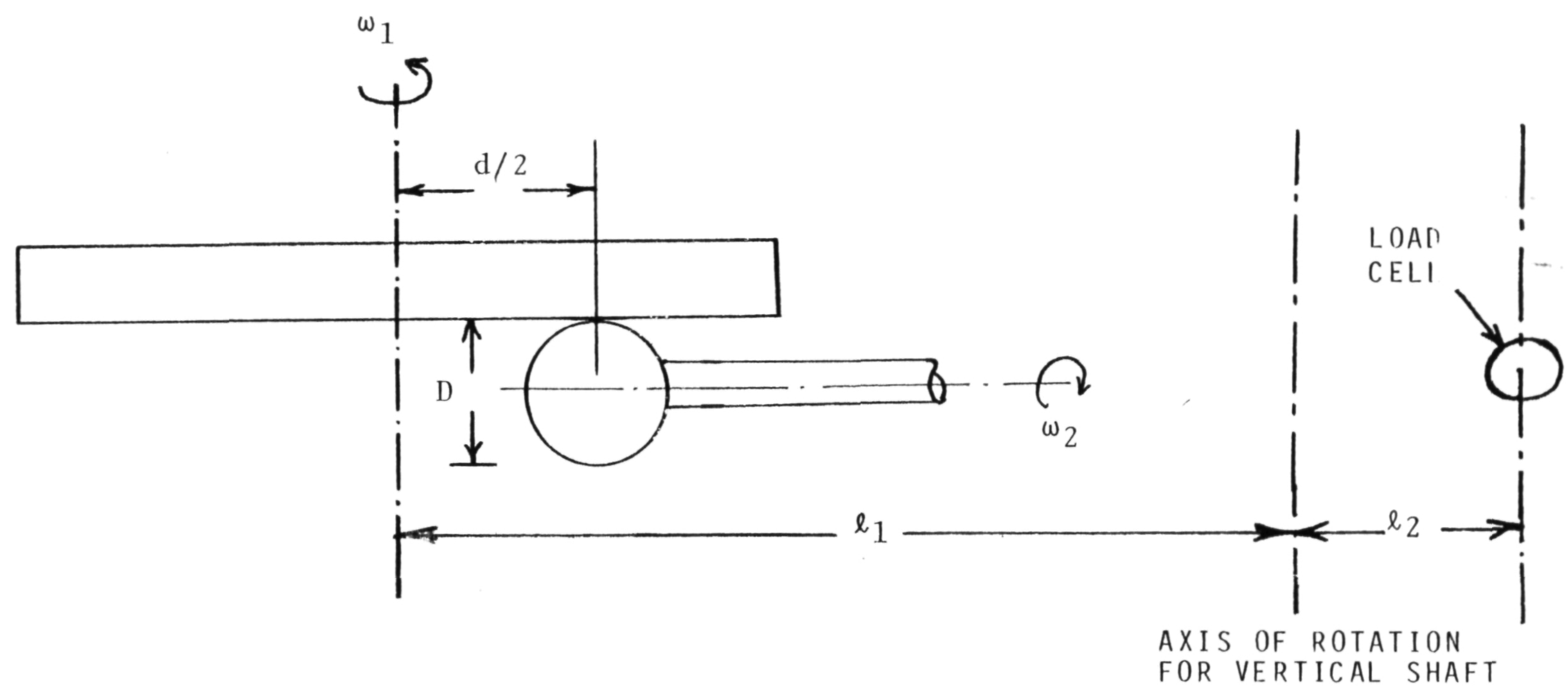


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FIGURE 6: CONTACT GEOMETRY FOR TRACTION FORCE AND SLIDE/ROLL RATIO MEASUREMENTS

slide/roll ratio curves can be obtained directly from the test rig. However, the traction force measured by the load cell is not the traction force at the contact. The traction force at the contact, T_C, is calculated from the traction load cell reading T_L using the relationship

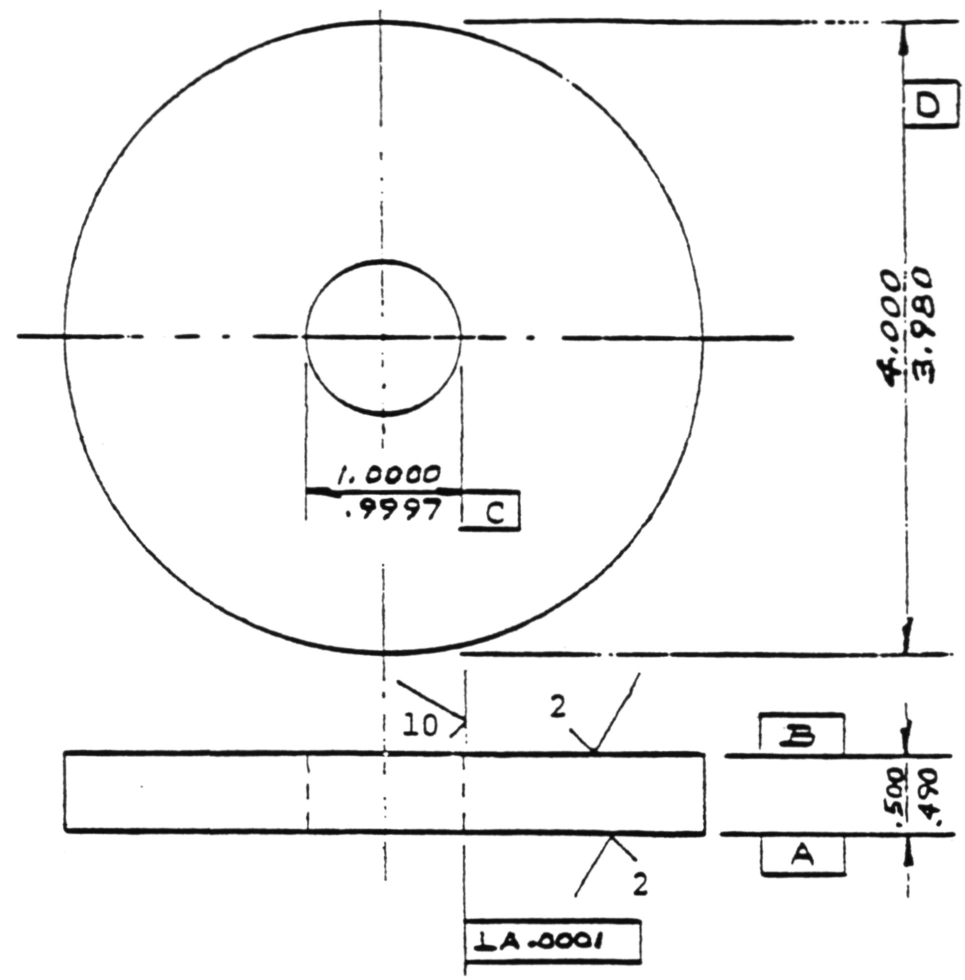
$$T_C = T_L \ell_2/(\ell_1 - 0.5d) \quad (2)$$

where ℓ_2 is the lever arm for the load cell and ℓ_1 is as shown in Figure 6. The value of ℓ_1 is 8.5 inches (215.9 mm) and ℓ_2 is 3.706 inches (94.13 mm). The lever arms are measured relative to the axis of rotation of the vertical shaft on which the rolling element motor is mounted. The calculation of the traction force at the contact is incorporated into the load cell calibration.

3.20 Test Specimens

3.21 Disc Specimens

Dimensions of disc specimens are shown in Figure 7. The disc materials used in the program were M50 steel, Norton NC132 hot pressed silicon nitride and Nielson's MS grade Partially Stabilized Zirconia (PSZ). The disc surfaces were ground such that the lay of surface roughness was circumferential. Surface roughness and hardness measurements for the disc specimens are sum-



Notes

1. Surfaces "A" and "B" to be flat within 2 to 4 waves per inch.
2. Surfaces "A" and "B" to be parallel 2 to 4 seconds of arc or better.
3. Surface finish "A" and "B" 2 µinch AA.
4. "C" and "D" to be concentric within 0.0002".
5. Diameter "D" within $\pm$ 0.001".
6. All dimensions are in inches.

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FIGURE 7: DRAWING OF DISC SPECIMEN

marized in Table 2. There was substantial variation in surface roughness from specimen to specimen. A photograph of an M50 disc specimen is shown in Figure 8.

3.22 Ball Specimens

Only ball specimens were used in this program. A new quill was designed to allow the use of interchangeable ball specimens. A drawing of the quill is shown in Figure 9. The ball specimens, shown in Figure 10, were standard 13/16" M50 and 440C bearing balls modified for use in the program. The ball-quill assembly is shown in Figures 8 and 11. A new ball was used for each test. 440C balls were used only during lubricant screening tests. Traction data for developing design guidelines was obtained using M50 balls. The M50 balls had a hardness of 60.4 Rc, average surface roughness of 0.8 μin and RMS roughness of 1.0 μin.

3.23 Lubricant Specimens

Lubricants tested in the program are listed in Table 3. The HAC composite lubricants were supplied gratis by Hughes Aircraft Company. All the other lubricants are commercially available.

The graphite lubricants are manufactured by Pure Carbon Company. Some properties of the lubricants taken from a Pure

TABLE 2: SURFACE ROUGHNESS AND HARDNESS MEASUREMENTS FOR DISC SPECIMENS

DISC MATERIAL	HARDNESS (Rc)	SURFACE ROUGHNESS (μin)	
		AVERAGE	RMS
M50 STEEL	60.4	1.4 - 6.3	1.9 - 7.9
NC 132 Si_3N_4	-	1.7 - 2.1	2.4 - 2.6
PSZ	-	3.7 - 9.9	5.2 -12.4

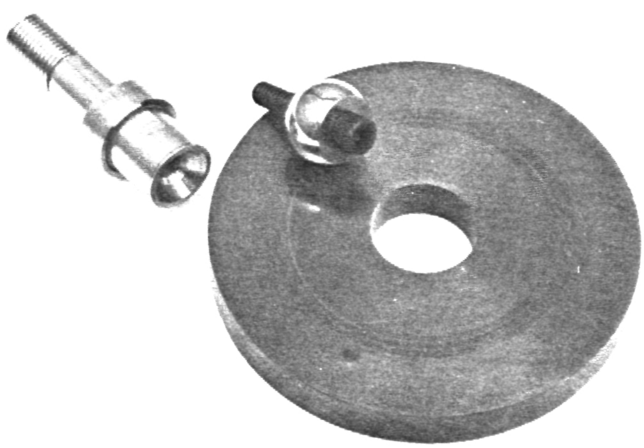

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FIGURE 8: M50 BALL AND DISC SPECIMENS

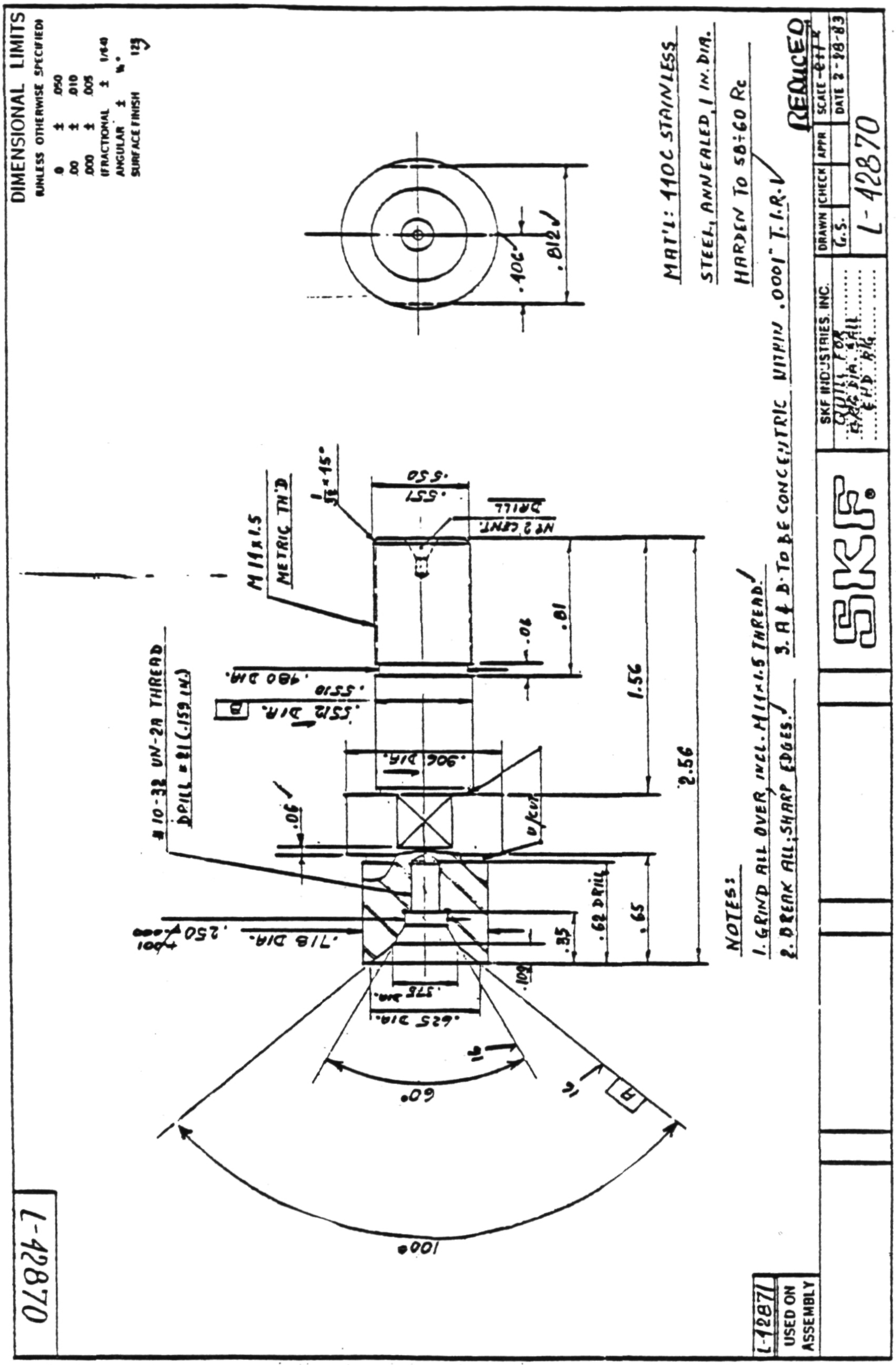

FIGURE 9: DRAWING OF QUILL

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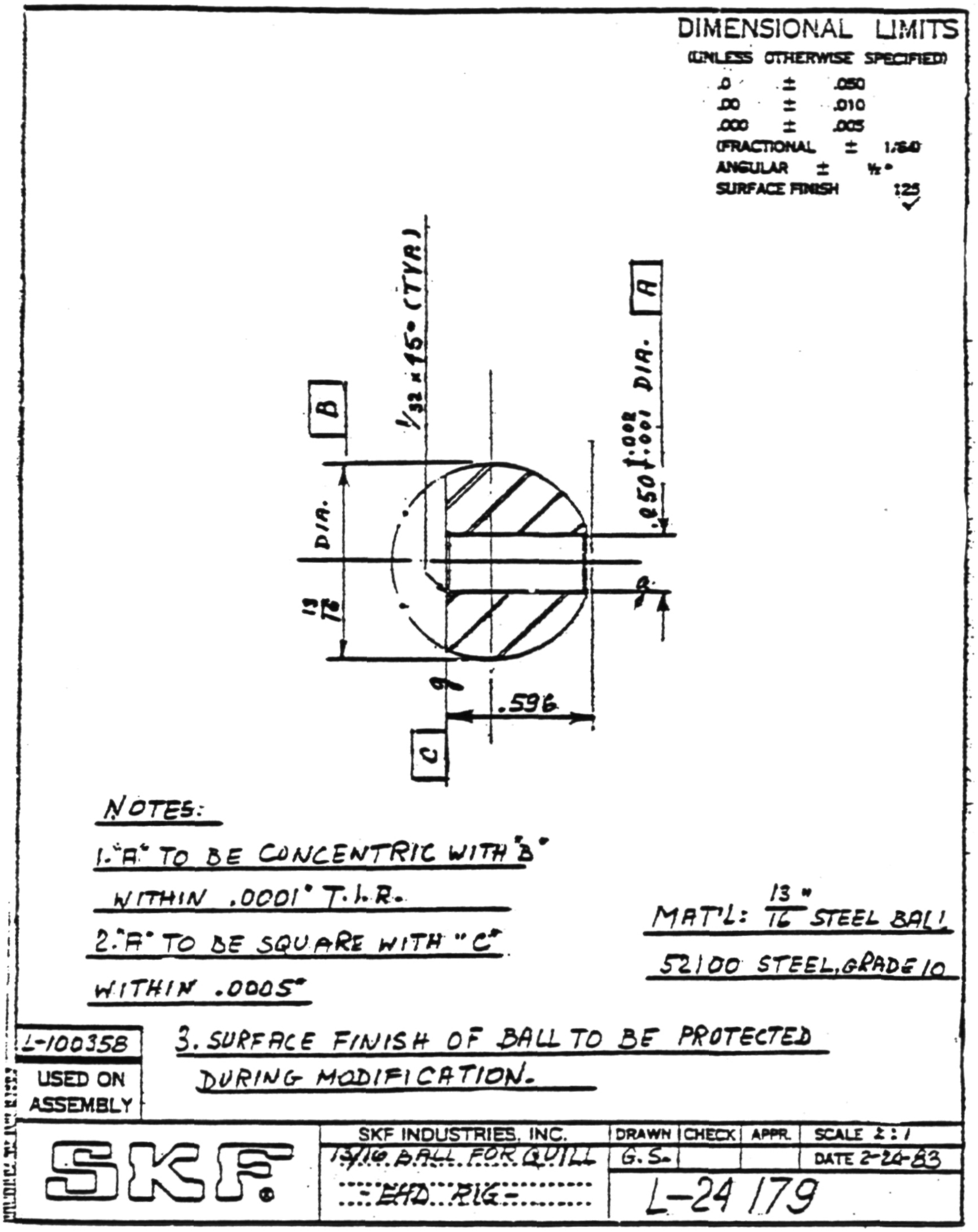

FIGURE 10: DRAWING OF MODIFIED BALL

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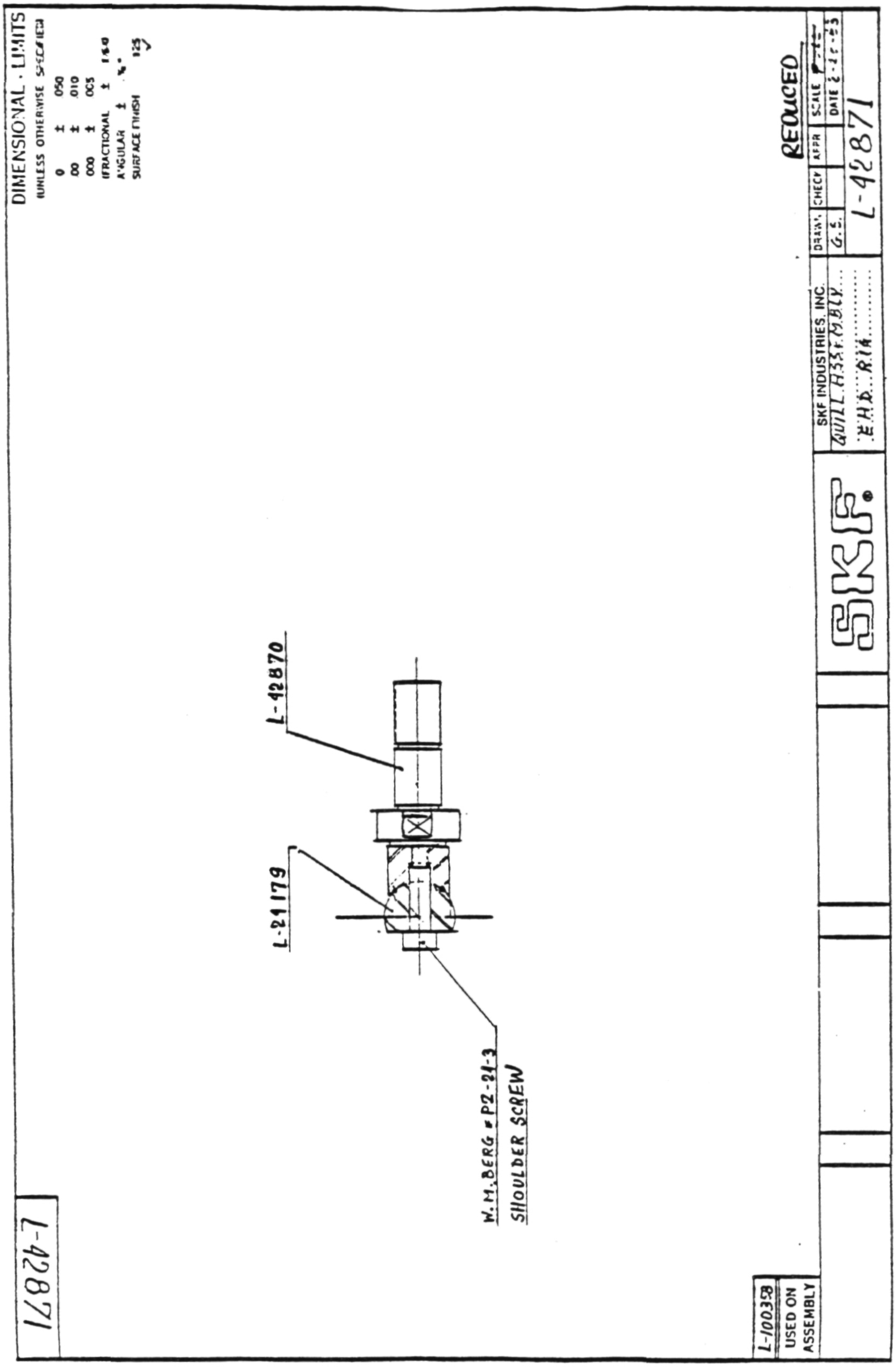

FIGURE 11: DRAWING OF BALL-QUILL ASSEMBLY

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TABLE 3: LUBRICANTS TESTED

P9	GRAPHITE
P03Ag	GRAPHITE
P3310	GRAPHITE
P2003	GRAPHITE
HAC1/T50F3	COMPOSITE
HAC1/T50F4	COMPOSITE
HAC2A/T50F4	COMPOSITE
POLYBON L	
POLYBON M	
ELECTROFILM SPRAY LUBRICANT (MoS_2 + GRAPHITE)	

Carbon brochure are listed in Table 4. Grade P9 is a low permeability, machinable carbon-graphite developed for sleeve bearings in submerged pumps [40]. Grade P03 carbons were developed for high speed face seals and have a high thermal conductivity, strength, abrasion resistance and oxidation resistance. P03 Ag is base P03 grade carbon impregnated with silver metal to promote thermal conductivity. Grade P2003 is a base P03 type material impregnated with relatively insoluable glass-like phosphates to improve its strength and permeability. In addition, the glass-like phosphates melt at high temperature and oxidize in preference to the graphite. The melt also helps to retain the graphite in the contact. The P2003 grade graphite tend to absorb moisture. The adsorbed moisture softens the additive and converts it into a gel which upon heating may exude and harden into a surface glaze. Information about grade P3310 graphite was not available. Composition of all these graphites is proprietary information.

Polybon L and M are polymeric composites made by Tribon Bearing Company. Polybon L (Grade T-0650) is a laminate fabricated with a high-strength PAN-based carbon fabric, 56% by volume, mated with a polyimide binder capable of continuous service at 600°F. Polybon M (Grade T-0454) is a self-lubricating molded polyimide with carbon and graphite fillers. This material can also be used up to 600°F.

TABLE 4: AVERAGE PHYSICAL PROPERTIES OF GRAPHITE LUBRICANTS

PROPERTY	LUBRICANT			
	P03AG	P9	P2003	P3310
APPARENT DENSITY (gm/cc)	2.40	1.75	2.	2.05
COMPRESSIVE STRENGTH (KSI)	29	21	33	30
TRANSVERSE STRENGTH (KSI)	11	7.5	12	11
TENSILE STRENGTH (KSI)	8.5	4	8.5	8.5
MODULUS OF ELASTICITY (PSI X 10^6)	2.4	2.1	2.2	2.6
TEMPERATURE LIMIT IN AIR (°F)	950	550	1250	1000
COEFFICIENT OF THERMAL EXPANSION (μIN/IN/°F)	2.5	2.4	2.7	2.3
THERMAL CONDUCTIVITY (BTU/HR/FT^2/°F/FT)	50	7	45	45

Electrofilm spray lubricant is a mix of MoS_2 and Graphite in an unspecified resin binder. The mixture is sprayed from a can onto the surface to be coated. The coating air dries in 30 minutes to allow handling of parts. The resin sets completely in 6 hours. M50 disc specimens were coated with Electrofilm to measure traction forces for coatings.

Development of the Hughes Aircraft Company composites was described in detail in Reference [33]. The composites are a polyimide resin reinforced with a 3-D weave of carbon fibers and impregnated with a lubricant. HAC1 contains a Westinghouse Compact as the lubricant while HAC2 contains Turbolube as the lubricant. Westinghouse Compact contains 80% WSe_2, 15% Ga and 5% In while Turbolube contains 60-75% MoS_2, 20-30% $MoSe_2$ and 5-7% WS_2 or WSe_2. The structure of the 3-D carbon weave is shown in Figure 12.

The graphite lubricants were supplied as 0.5 inch long cylinder of 0.5 inch diameter. The composite lubricants were supplied as rectangular slabs. The dimensions of the specimens as used in tests are shown in Figure 13, along with a photograph of the specimen holder. The rectangular specimens were 0.25 inches thick.

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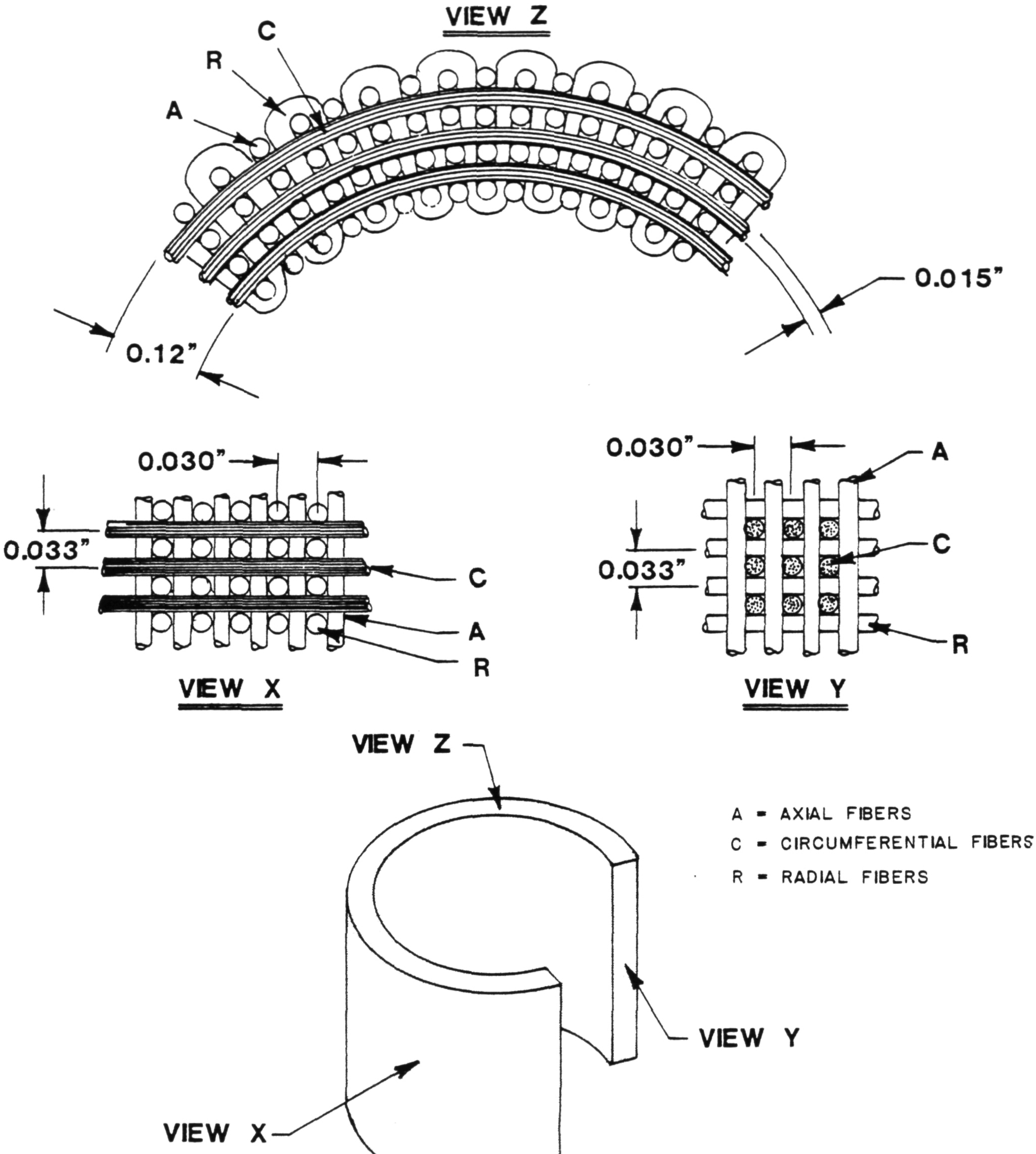
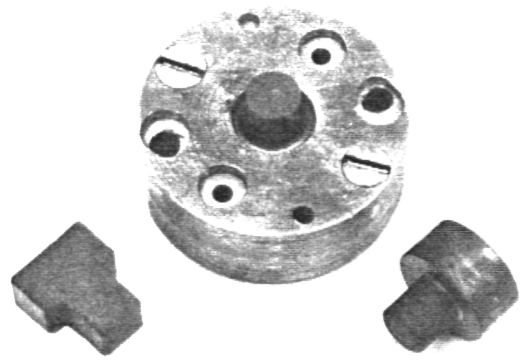


FIGURE 12: 3-D WEAVE USED IN HAC1 AND HAC2 COMPOSITES (FROM REF. [41])



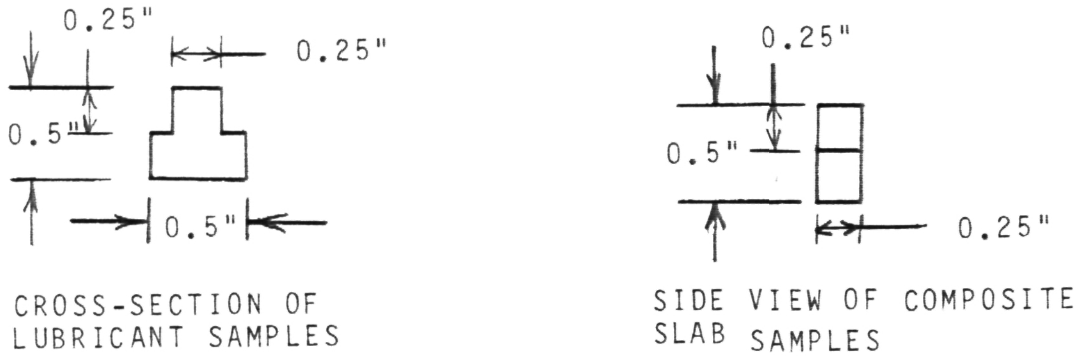


FIGURE 13: LUBRICANT SPECIMENS AND LUBRICANT HOLDER

3.30 Test Matrix

The contact load and rolling speeds selected for testing are shown in Table 5. The test matrix was selected to cover the range of operating conditions anticipated for rolling element bearings in advanced adiabatic diesel engines.

3.40 Development of Test Procedure

The test procedure used at the beginning of the test program was based on the experience gained during traction measurements for oil lubricated contact. Grade P-9 graphite was used as the lubricant for developing the test procedure for solid lubricants. The initial test procedure was as follows:

(1) Burnish lubricant on the disc surface for 15 minutes using a burnishing load of 6 lbs. The disc speed during burnishing was the same as that to be used for traction force measurements.

(2) Spin the ball at the required speed for pure rolling at the contact and bring the ball in contact with the disc at the desired load.

(3) Generate the traction curve by varying the disc speed to introduce sliding in the contact.

(4) Calibrate the electronic circuit used to calculate the slide/roll ratio.

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TABLE 5: TEST MATRIX

AVERAGE SURFACE SPEED (IPS)	MAXIMUM HERTZ CONTACT STRESS (KSI)		
	150	250	300
150	X	X	X
300	X	X	X

In addition to the traction curves, traction force was measured as a function of time at a fixed slide/roll ratio to test the durability of the burnished lubricant film under varying amounts of sliding. The traction vs. time curves were usually run for 15 minutes.

The early testing revealed a number of problems with the test procedure. The lubricant supply proved inadequate for tests run at contact stresses other than 150 ksi. A series of tests were run using different burnishing loads and burnishing time to determine the best combination for adequate transfer of lubricant from the compact to the disc surface. A burnishing load of 24 lbs for 30 minutes was chosen. Subsequent testing showed insufficient lubricant for tests at 300 ips rolling speed and 300 ksi contact stress. It was found necessary to burnish the ball surface also to provide sufficient lubricant. The ball surface was burnished for 30 minutes using a burnishing load of 12 lbs.

The traction force level proved to be an inadequate criteria for determining the success or failure of the lubricant. The condition of contact surfaces after the test was found to be just as important as the traction forces. The ball specimens were examined in a Scanning Electron Microscope (SEM) after each test; the discs were too large to fit in the SEM. However, since the

disc surface is lubricated during the test and the ball surface is not, the condition of the ball surface was considered a good measure of the wear protection provided by the lubricant. Smaller disc specimens are being acquired to allow SEM examination of disc surfaces for the high temperature tests in the second phase of the program. The wear of contact surfaces also necessitated the use of a fresh ball specimen and a new track on the disc for each test.

The test procedure was standardized as follows:

(1) Burnish lubricant on the ball and disc surfaces for 30 minutes to provide a supply of lubricant on both surfaces. The lubricant burnishing load was 12 lbs for the ball and 24 lbs for the disc surface.

(2) Bring the ball and the disc into contact at the desired load with surface speeds set for pure rolling at the contact. The contact is lubricated during the test by burnishing the lubricant on the disc surface. The ball surface is not lubricated during the test.

(3) Generate the traction curve or traction vs. time curve.

(4) Examine the ball surface in the SEM.

(5) Calibrate the electronic circuit used to calculate the slide/roll ratio.

3.50 Test Rig Modifications

A number of deficiencies in the test rig were revealed during the development of the test procedure. The high speed spindle for the ball specimen, designed for speeds up to 42,000 RPM, could not supply enough power to drive the contact at contact stresses of 300 ksi because it was being run at or near the minimum recommended speeds (10-15% of maximum speed). The problem was fixed by changing the speed controller to increase the power available at the speeds used in the test program (3500 to 7000 rpm). The changes enabled the tests to be run for positive slide/roll ratio for all load and speed combinations in the test matrix but the motor could not supply power for running with negative slide/roll ratios (ball surface speed faster than disc surface speed) at 300 ksi.

The maximum traction force levels were higher than expected. As a result, the readings often exceeded the range of the traction load cell and the load used to clamp the load cell to its fixture. A new load cell (0 to 100 lbs range) was purchased to replace the original load cell and the clamping load was

increased to prevent the movement of the load cell in its fixture.

Traction measurement tests with oil lubricated tests had shown that the normal load at the ball-disc contact was higher than that indicated by the load cell and that the contact load varied over time. The variation was significant at low normal loads. The ball specimens and the quill designed for the solid lubrication program were used to run tests with oil lubricated contact in order to assess the dynamic load variation using interferograms. For tests run with SANTOTRAC 50 traction fluid at 150 ips rolling speed and a 10 lb normal load, the contact load measured using 17 interferograms averaged 12 lbs with a variation of ±3 lbs. Oscilloscope traces of the load cell signal showed a load variation of similar magnitude at a frequency of 3 cps. The load variation for the quill designed for solid lubricated tests was lower than that observed for quills used for oil lubricated tests. The lower variation was attributed to the low runout (25 μin) of the new quills. The situation was considered acceptable as long as the load variation for solid lubricated contacts did not exceed that observed for oil lubricated contacts. There was, however, no way of assessing the contact load variation for solid lubricated contacts other than the value indicated by the load cell. Further investigation using oil lubricated tests was ruled out because of time constraints.

3.60 Program Test Plan

The test program was divided into two phases. In the first phase, the 10 lubricants listed in Table 3 were tested to select the four most promising lubricants. All the screening tests were run using 440C steel balls and M50 steel discs. In the second phase, the lubricants selected during the screening tests were tested using M50 balls, and all the disc materials used in the program: M50 steel, NC132 silicon nitride and Nielsen's MS Grade PSZ. Tests run during both phases of the program were used for generating qualitative guidelines for the design of solid lubricated contacts. The screening tests also provided vital experience for interpreting the results of the tests.

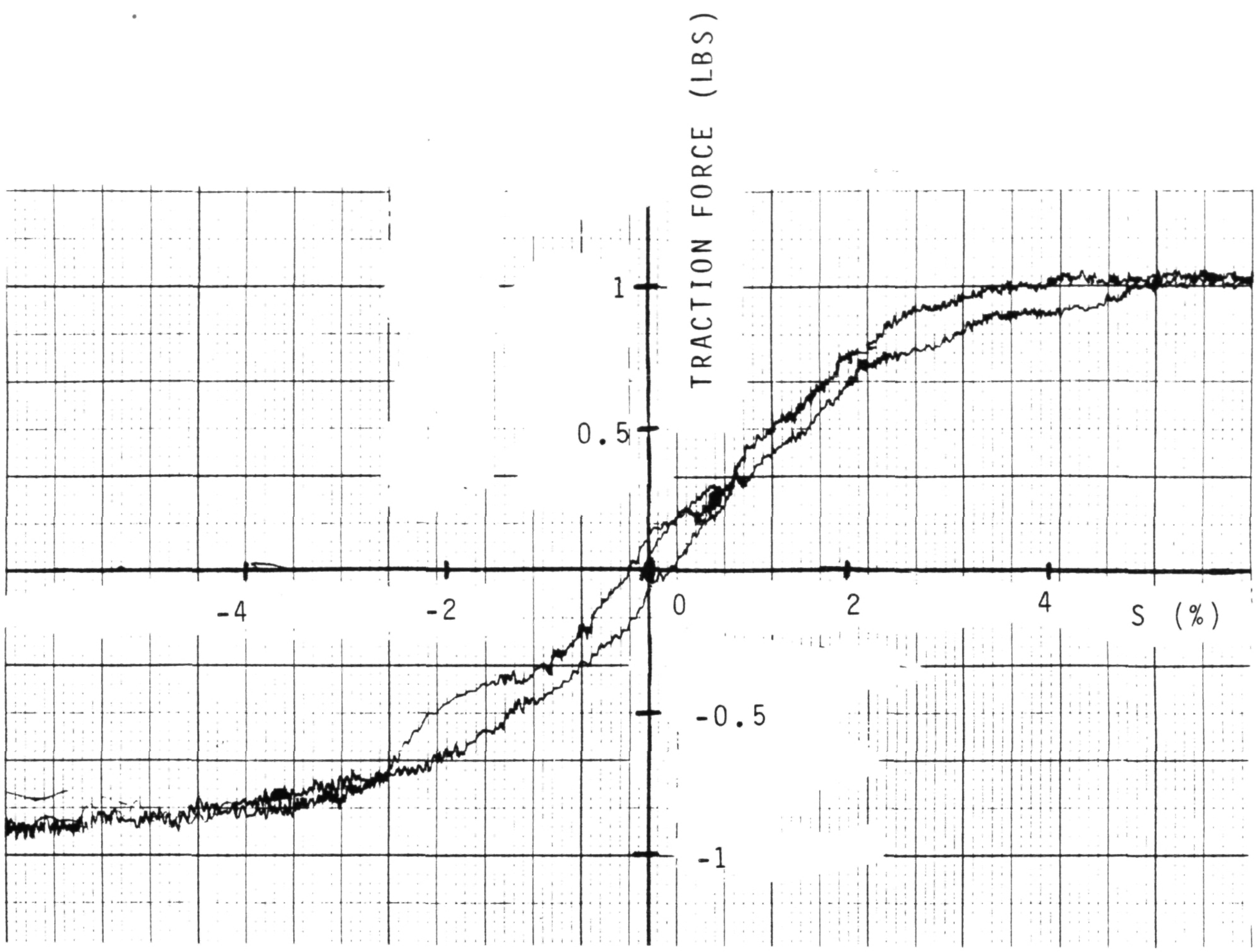
4.0 RESULTS OF SCREENING TESTS

All the lubricant screening tests were run using 440C balls and M50 discs. The runout averaged 25 μin for ball specimens and 150 μin for disc specimens. The results for all the lubricants are discussed below.

4.10 Electrofilm Spray Lubricant

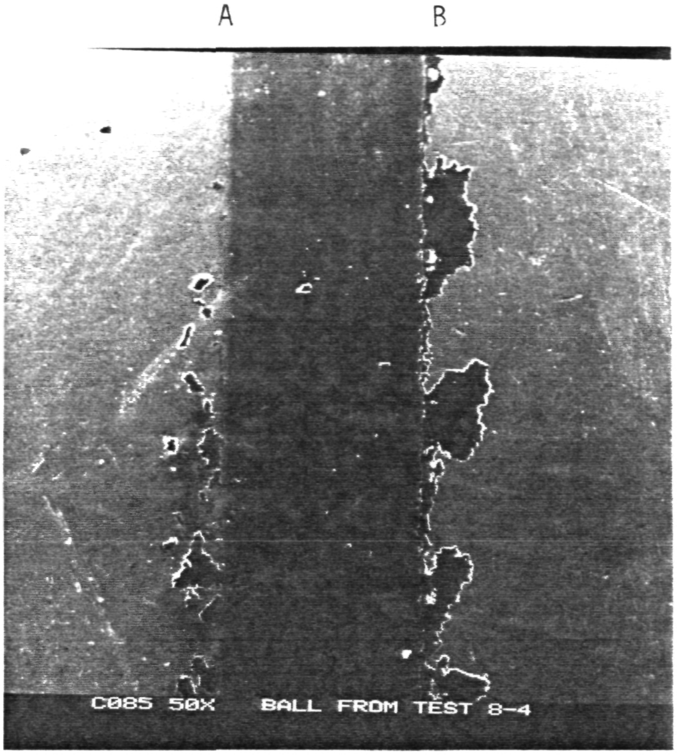
Electrofilm spray lubricant was the only lubricant coating tested in the current test progam. The disc specimens were first cleaned with acetone. Electrofilm was sprayed on the disc surfaces and allowed to dry for a minimum of 6 hours. Both traction force vs. slide/roll ratio curves and traction force vs. time curves were generated.

A traction vs. slide/roll ratio curve at 150 ips rolling speed and 150 ksi contact stress is shown in Figure 14. The traction force rises to a value of 1 lb and then levels off at a slide/roll ratio of approximately 4%. This curve is typical of the traction curves obtained for all the lubricants. SEM pictures of the ball specimen are shown in Figure 15. A 50X picture of the wear track is shown in Figure 15a while 250X pictures of the edges of the track are shown in Figures 15b and 15c. Figure 15d shows a 1000X picture of surface in the wear track and



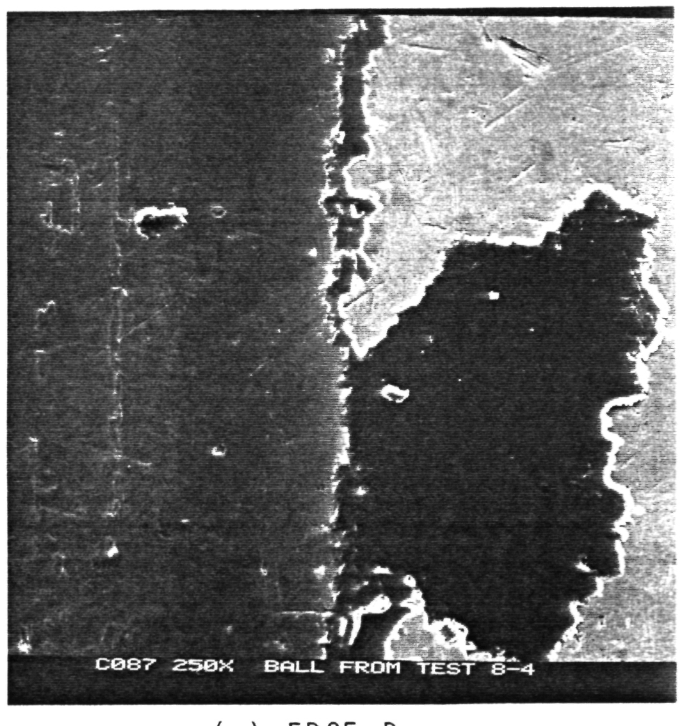
LUBRICANT: ELECTROFILM SPRAY
NORMAL LOAD: 12 LB
CONTACT STRESS: 150 KSI
ROLLING VELOCITY: 150 IPS

FIGURE 14: TRACTION CURVE FOR ELECTROFILM SPRAY LUBRICANT



(a) WEAR TRACK

(b) EDGE A



(c) EDGE B

FIGURE 15: SEM PICTURES OF BALL WEAR TRACK FOR TEST IN FIGURE 14

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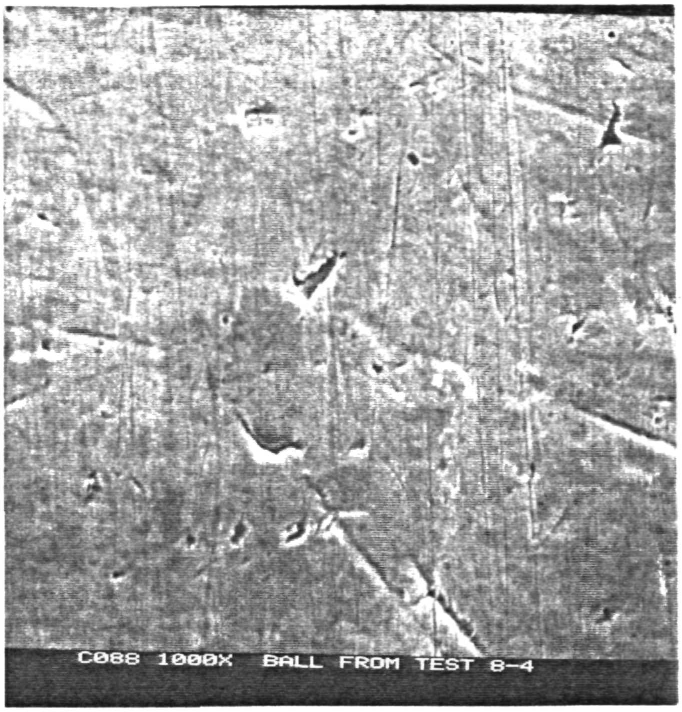
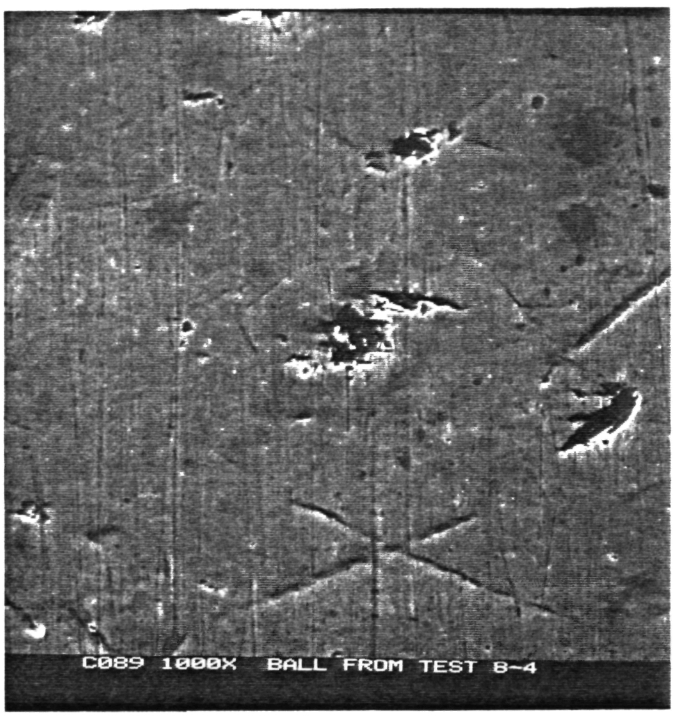


(d) WEAR TRACK



(e) VIRGIN SURFACE

FIGURE 15: SEM PICTURES OF BALL WEAR TRACK FOR TEST IN FIGURE 14 (CONTINUED)

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Figure 15e a 1000X picture of virgin ball surface outside the track. The location of edges A and B shown in Figures 15b and 15c, respectively, are as shown in Figure 16. Figure 15 shows that a layer of lubricant was transferred from the disc to the ball and that the lubricant supply in the contact was sufficient to protect the surface for the duration of the test. Values of the limiting traction force for other loads and speeds are listed in Table 6; the limiting traction force is defined as the level at which the traction force levels off in a traction curve.

A traction vs. time curve for Electrofilm is shown in Figure 17. The curve was obtained at pure rolling (slide/roll ratio equal to zero) at a speed of 150 ips and a contact stress of 150 ksi. SEM pictures of the ball specimen are shown in Figure 18. The arrangement of SEM pictures for all ball specimens is the same as in Figure 15. A traction vs. time curve obtained at an average rolling speed of 150 ips and a contact stress of 150 ksi at pure rolling for about 7 minutes and then a slide/roll ratio of 2% for 5 minutes is shown in Figure 19. SEM pictures of ball specimens (Figure 20) showed that the ball surface was heavily damaged. The lubricant coating was unable to protect the surfaces in the presence of sliding at the contact.

Tests with Electrofilm showed that while lubricant coatings do protect the contact surfaces in pure rolling contacts, pre-

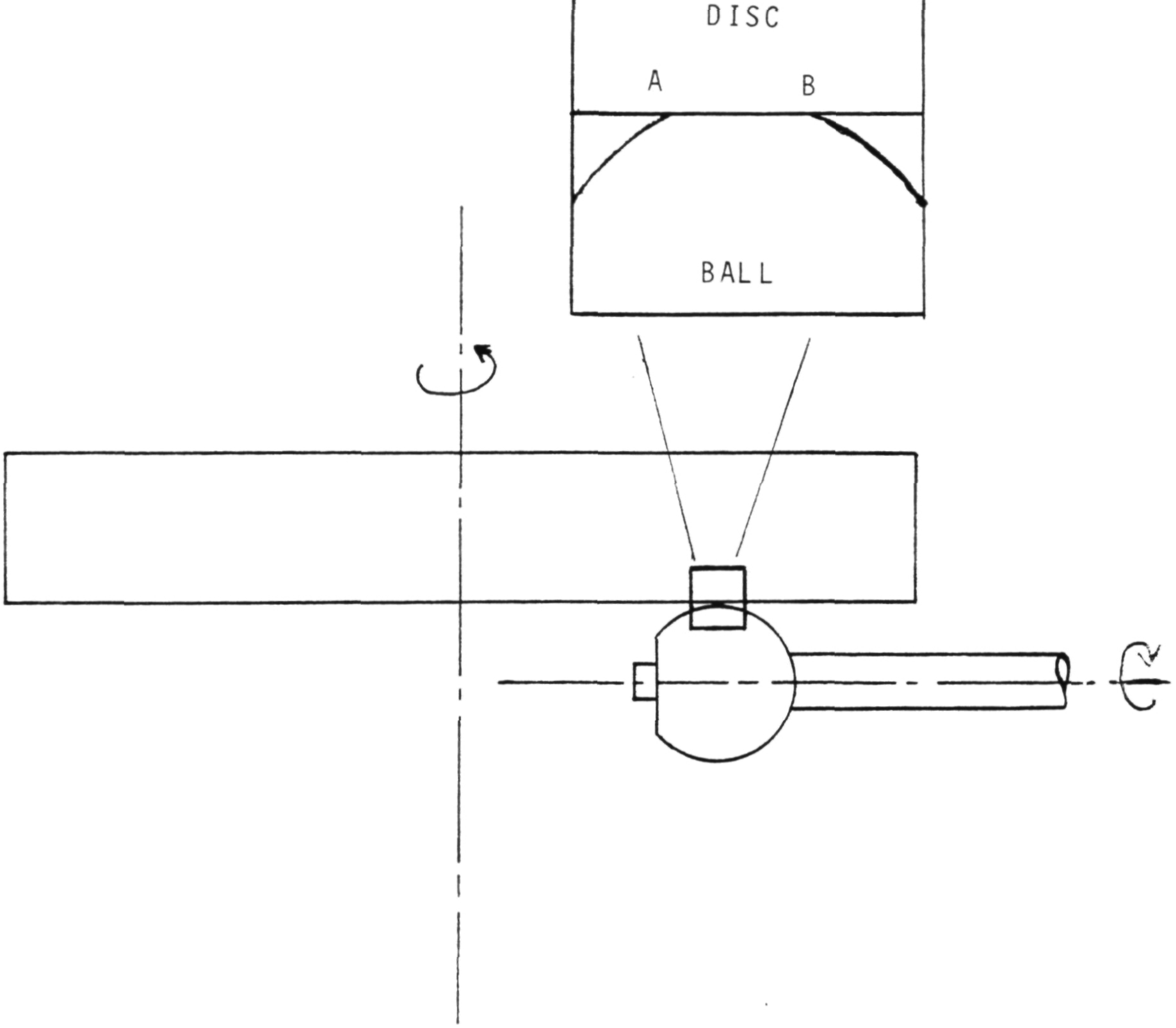
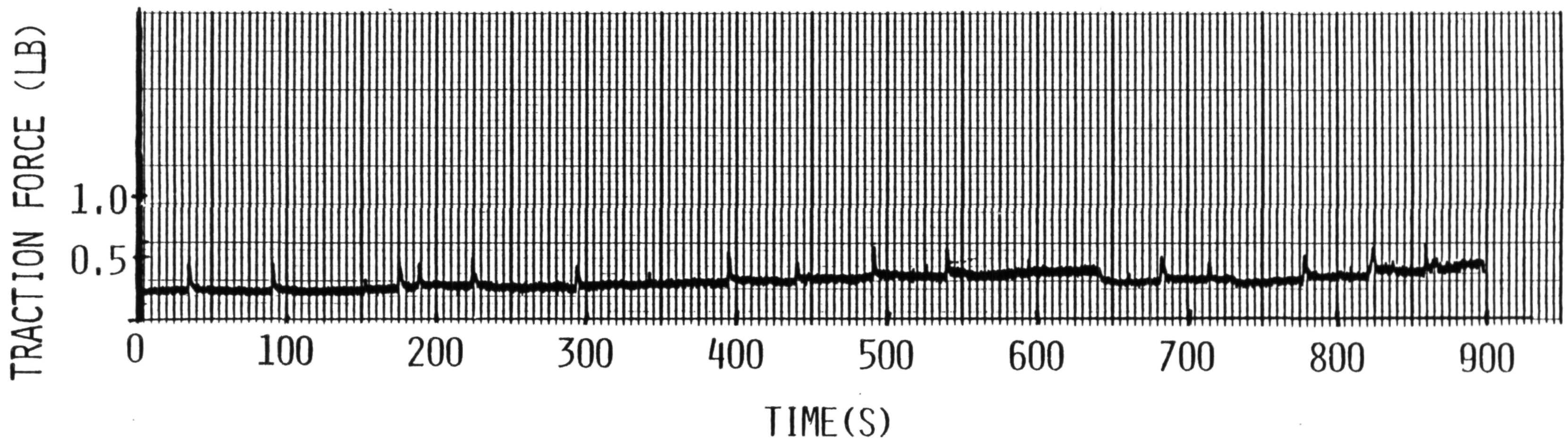


FIGURE 16: LOCATION OF EDGES A AND B IN SEM.
PICTURES OF BALL WEAR TRACKS.

TABLE 6: LIMITING TRACTION FORCE FOR ELECTROFILM SPRAY LUBRICANT

ROLLING SPEED (IPS)	NORMAL LOAD (LBS)	MAXIMUM HERTZ CONTACT STRESS (KSI)	LIMITING TRACTION FORCE (LB)
150	12	150	1.0
300	12	150	1.2
150	54	250	3.1
300	54	250	7.7
150	93	300	7.7
300	93	300	9.6



LUBRICANT: ELECTROFILM SPRAY
NORMAL LOAD: 12 LB
CONTACT STRESS: 150 KSI
ROLLING VELOCITY: 150 IPS
SLIDE/ROLL RATIO: 0%

FIGURE 17: TRACTION VS. TIME FOR ELECTROFILM SPRAY LUBRICANT AT ZERO SLIDE/ROLL RATIO

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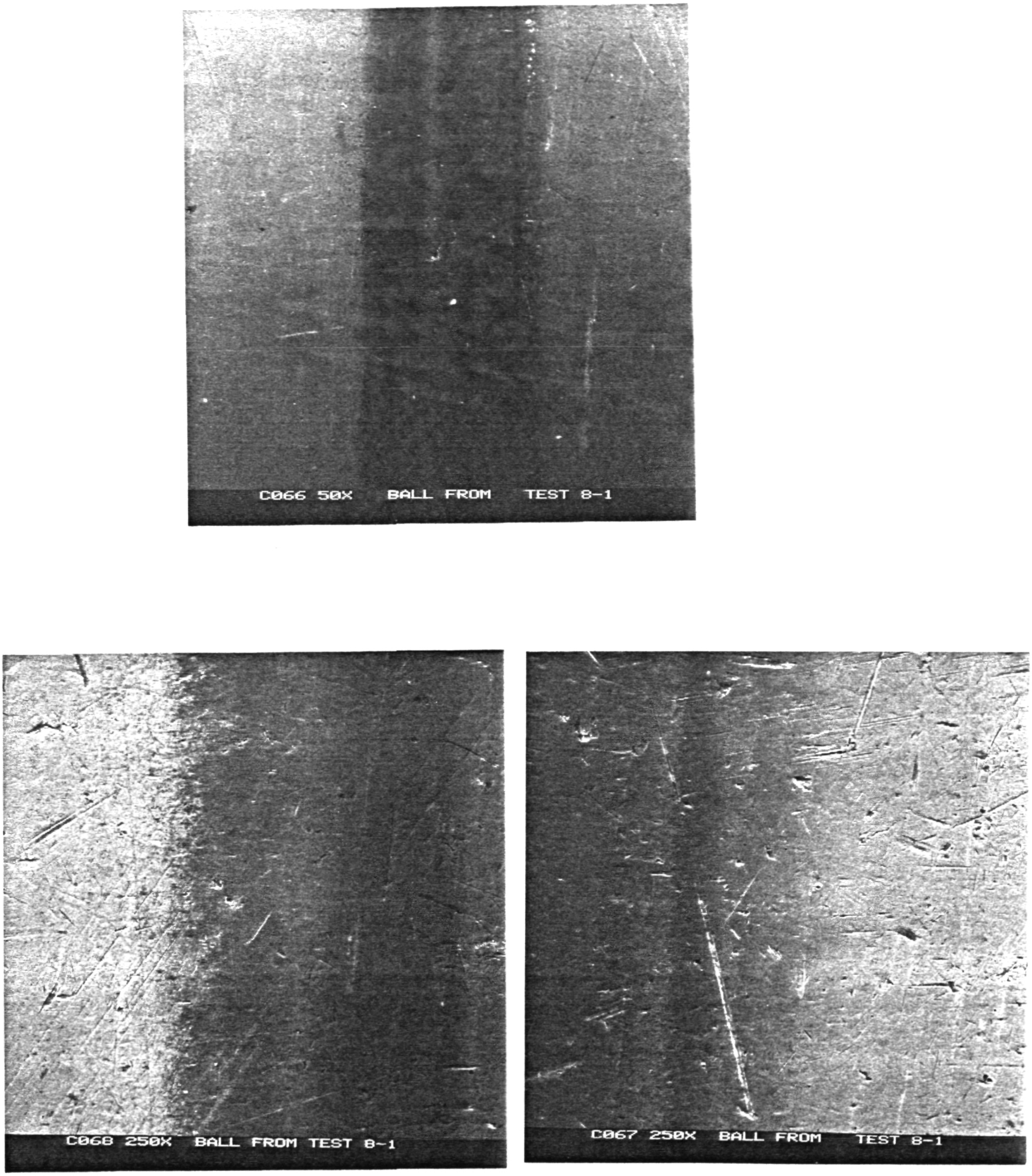

FIGURE 18: SEM PICTURES OF BALL WEAR TRACK FOR TEST IN FIGURE 17

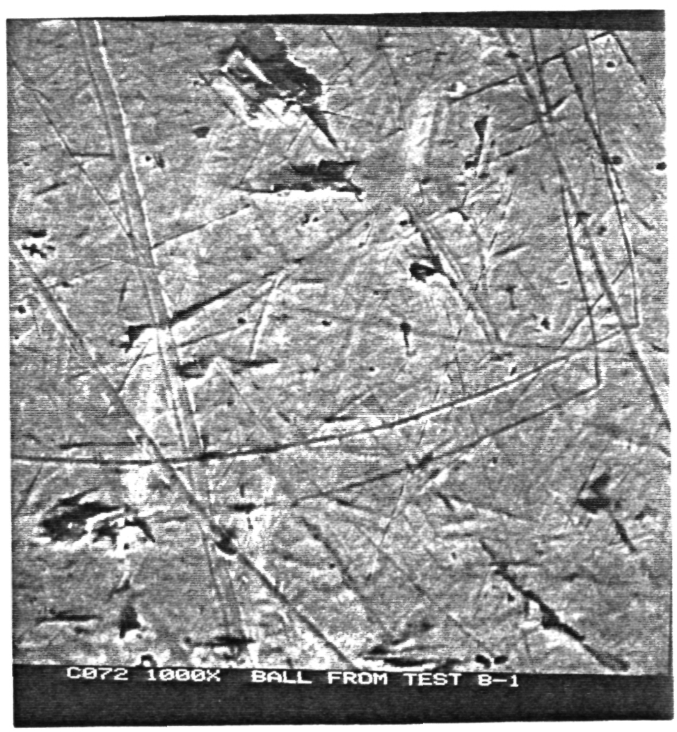

FIGURE 18: SEM PICTURES OF BALL WEAR TRACK FOR TEST IN FIGURE 17 (CONTINUED)

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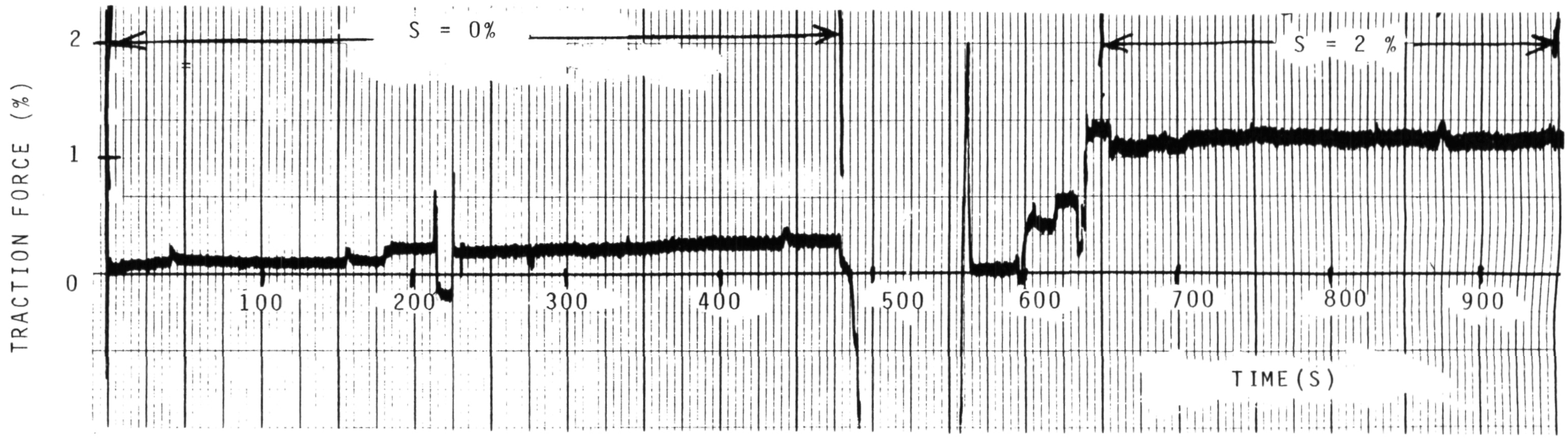

LUBRICANT: ELECTROFILM SPRAY
NORMAL LOAD: 12 LBS
CONTACT STRESS: 150 KSI
ROLLING VELOCITY: 150 IPS
SLIDE/ROLL RATIO: 0% AND 2%

FIGURE 19: TRACTION VS. TIME FOR ELECTROFILM SPRAY LUBRICANT AT SLIDE/ROLL RATIOS OF 0% AND 2%

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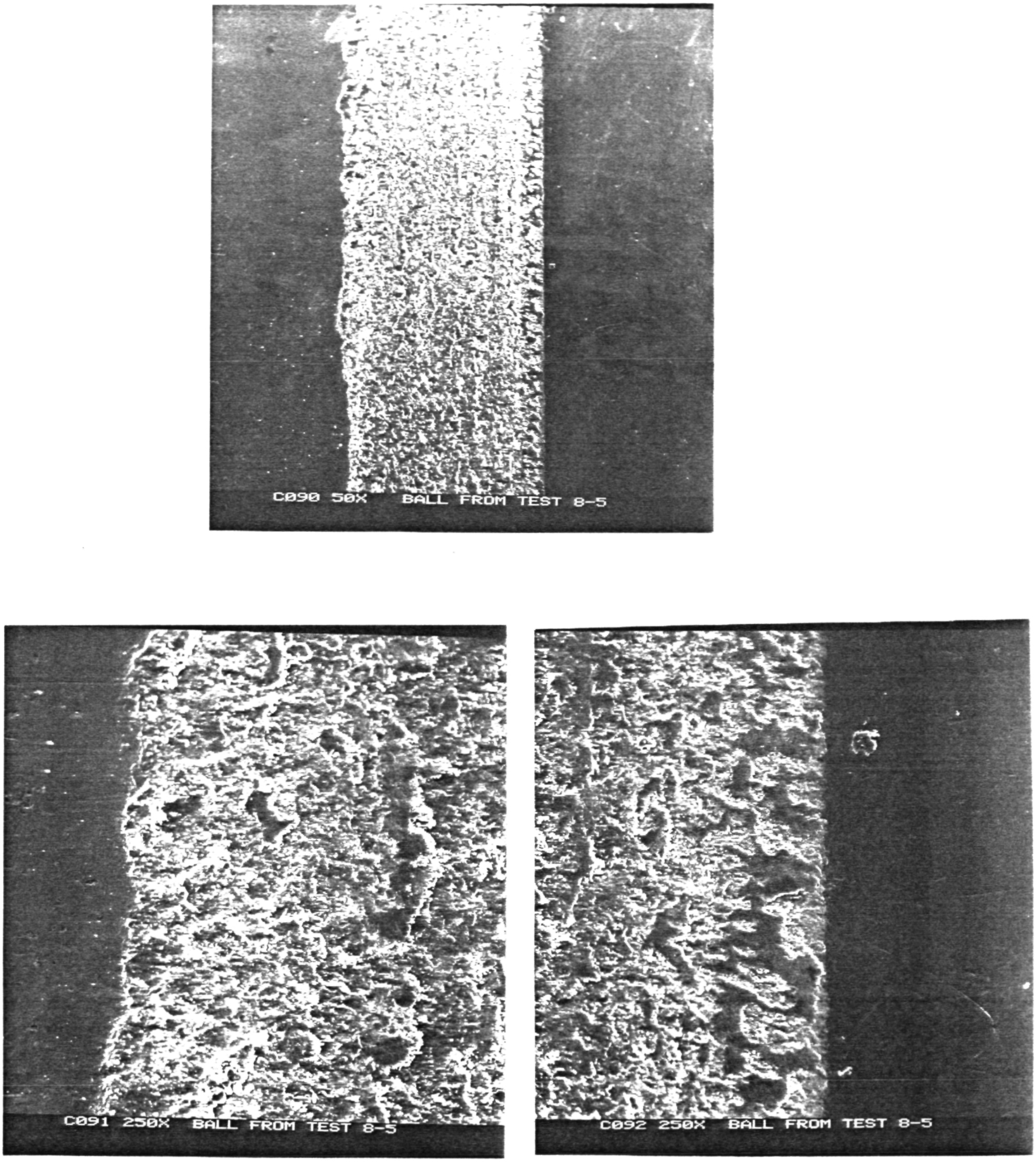

FIGURE 20: SEM PICTURES OF BALL WEAR TRACK
FROM TEST IN FIGURE 19

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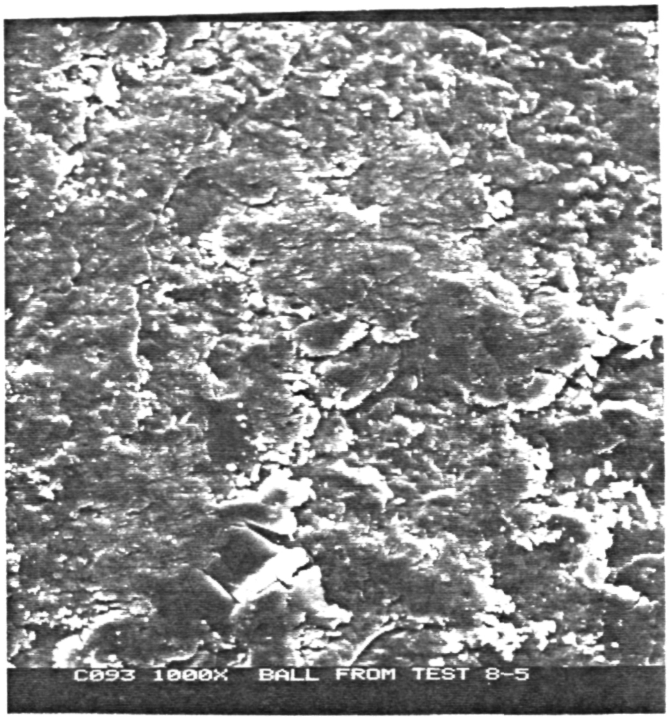

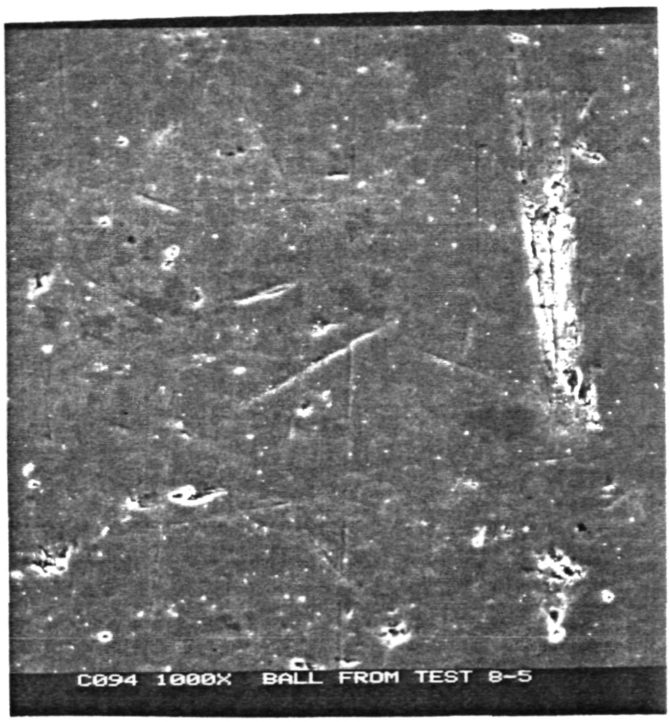

FIGURE 20: SEM PICTURES OF BALL WEAR TRACK FROM TEST IN FIGURE 19 (CONTINUED)

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sence of sliding depletes the lubricant supply rapidly. Since most rolling bearing applications have sliding at the rolling element race contacts, lubricant coatings on bearing races or rolling elements are not sufficient by themselves for adequate lubrication. Lubricant supply at the contact must be replenished in some way.

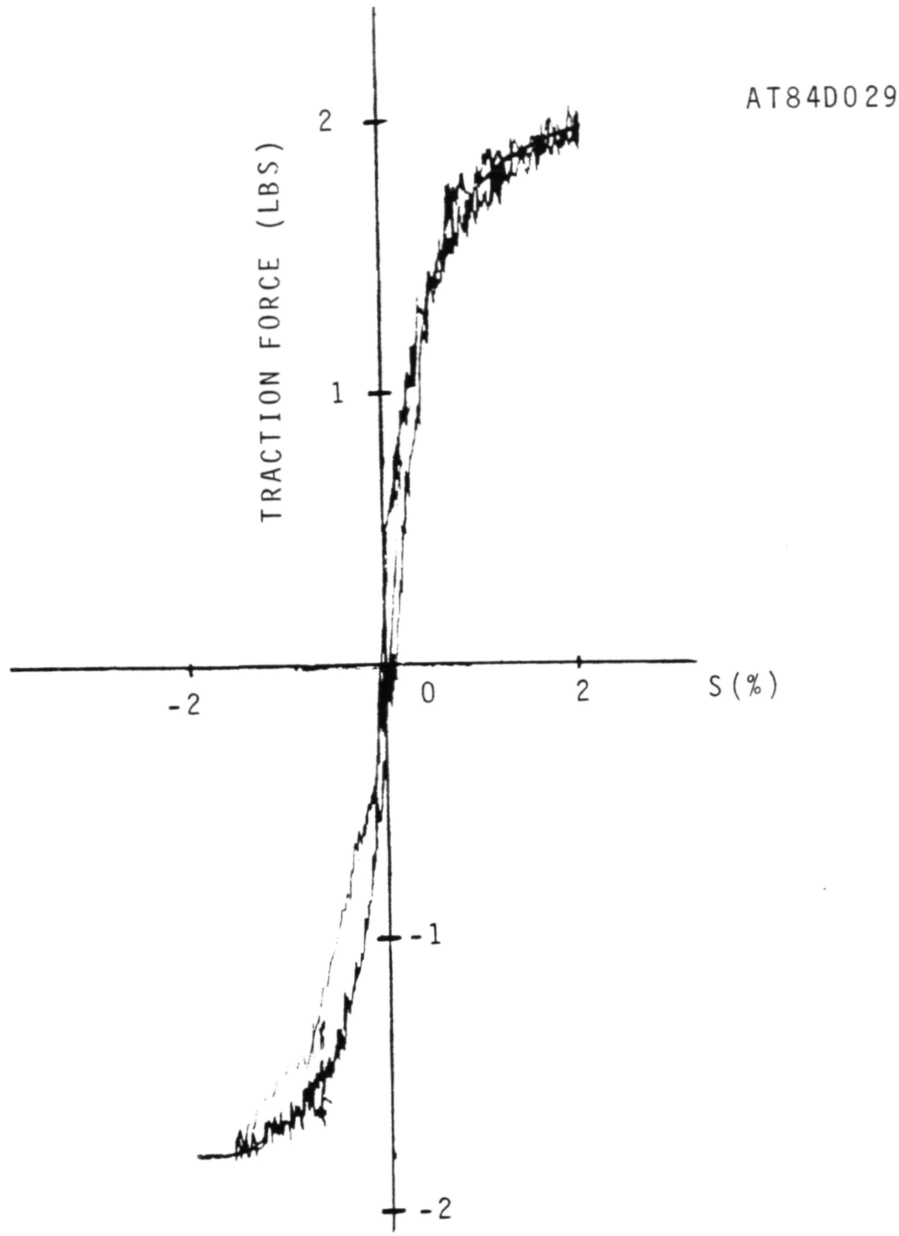
4.20 Graphite Lubricants

The four graphite lubricants listed in Table 3 were tested for effectiveness as lubricants to be used as bearing cage materials. The 250 ksi contact stress value in the test matrix in Table 5 was not used for the screening tests. Grade P2003 and P3310 have been used as high temperature cage materials in bearing tests at SKF Industries.

Grade P9 graphite proved to be the poorest of the four graphites in protecting the contact surfaces from wear and yielded the highest levels of traction forces. This was to be expected because P9 does not have any impregnants to improve its ability to lubricate. The limiting traction force levels for all the graphites are listed in Table 7. Figures 21, 23, and 25 show traction curves for P3310, P2003 and P03Ag graphites, respectively, at a rolling speed of 300 ips and a contact stress of 150 ksi. SEM pictures of ball wear tracks for the three trac-

TABLE 7: LIMITING TRACTION FORCE FOR GRAPHITE LUBRICANTS

LUBRICANT	ROLLING SPEED (IPS)	NORMAL LOAD (LB)	MAXIMUM HERTZ CONTACT STRESS (KSI)	LIMITING TRACTION FORCE (LB)
P9	150	12	150	1.9
	300	12	150	2.1
	150	93	300	19.7
	300	93	300	22.3
P3310	150	12	150	0.8
	300	12	150	1.9
	150	93	300	14.5
	300	93	300	19.4
P03AG	150	12	150	1.1
	300	12	150	1.3
	150	93	300	13.6
	300	93	300	18.0
P2003	150	12	150	3.0
	300	12	150	3.3
	150	93	300	20.0
	300	93	300	20.4



LUBRICANT: P3310
NORMAL LOAD: 12 LBS
CONTACT STRESS: 150 KSI
ROLLING SPEED: 300 IPS

FIGURE 21: TRACTION CURVE FOR P3310 GRAPHITE AT 300 IPS ROLLING SPEED AND 150 KSI CONTACT STRESS

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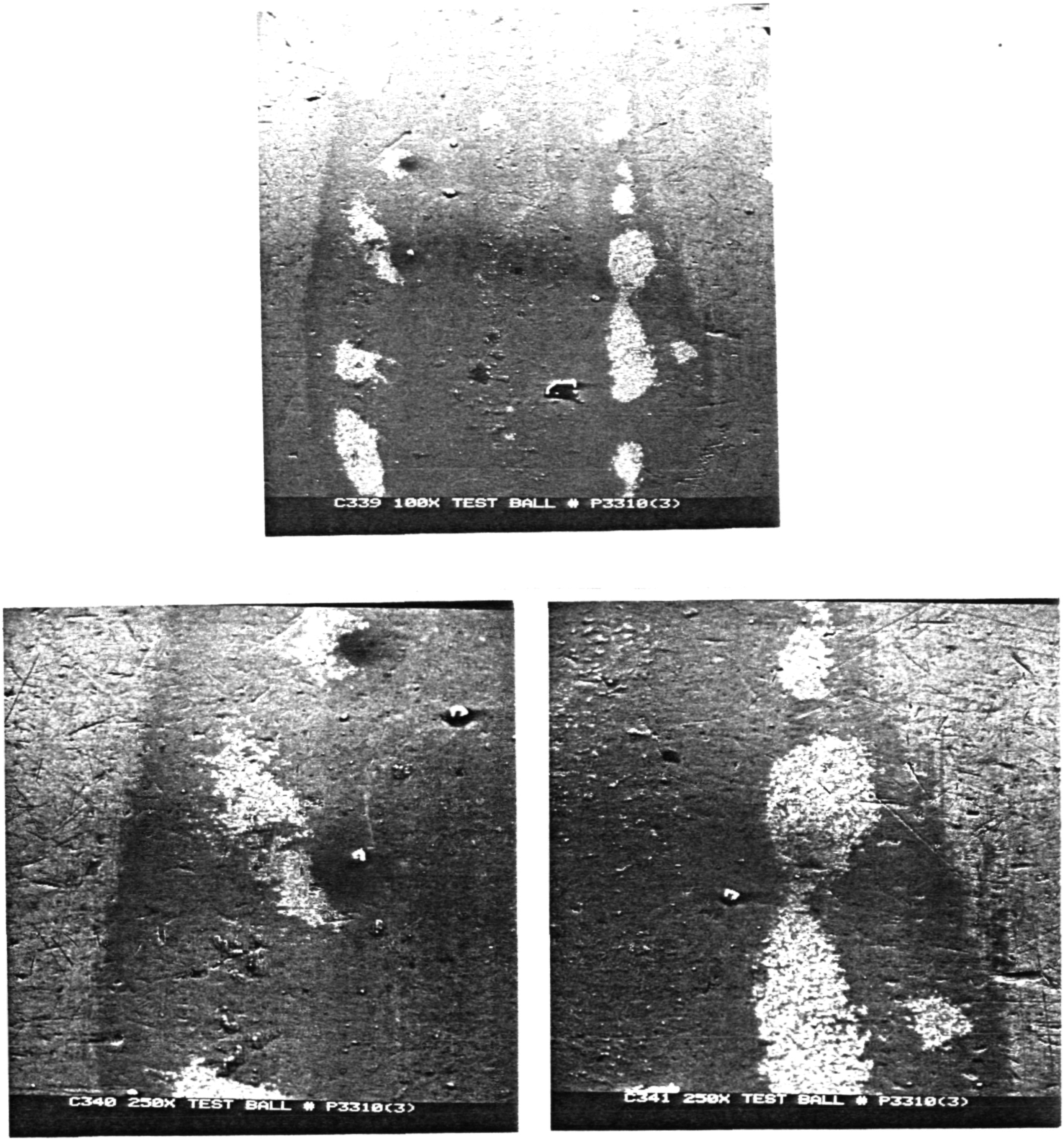


FIGURE 22: SEM PICTURES OF BALL WEAR TRACK FOR THE TEST IN FIGURE 21

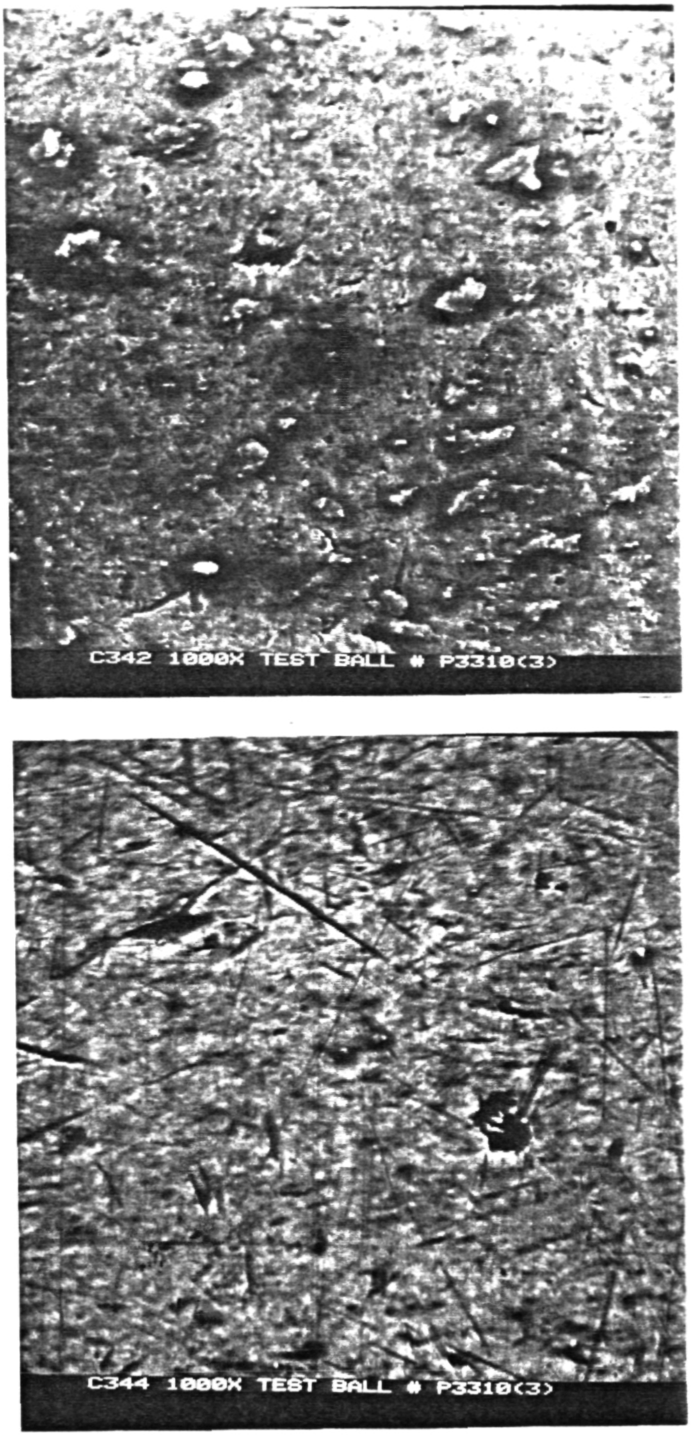
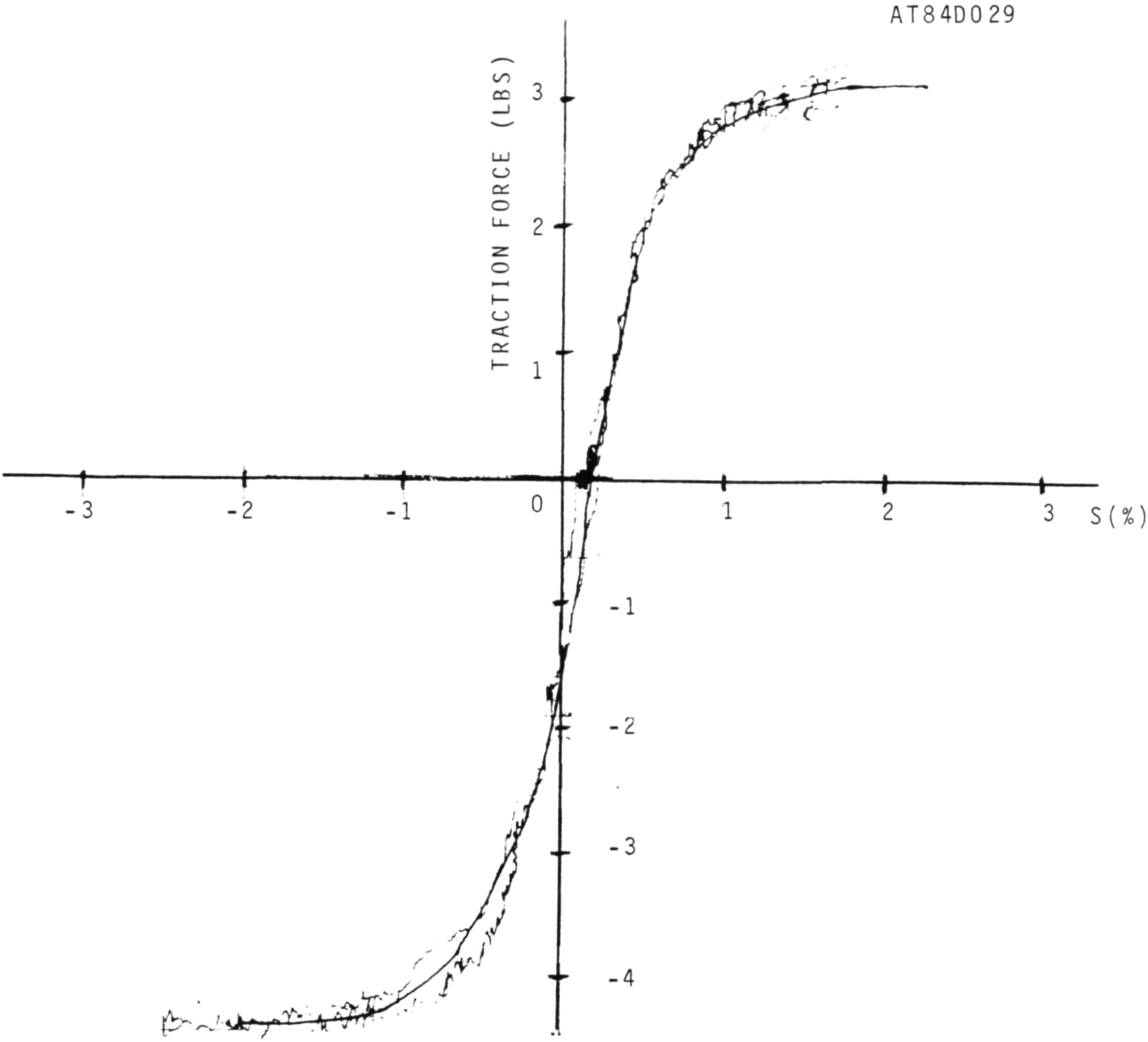


FIGURE 22: SEM PICTURES OF BALL WEAR TRACK FOR THE TEST IN FIGURE 21 (CONTINUED)



LUBRICANT: P2003
NORMAL LOAD: 12 LBS
CONTACT STRESS: 150 KSI
ROLLING SPEED: 300 IPS

FIGURE 23: TRACTION CURVE FOR P2003 GRAPHITE AT 300 IPS ROLLING SPEED AND 150 KSI CONTACT STRESS

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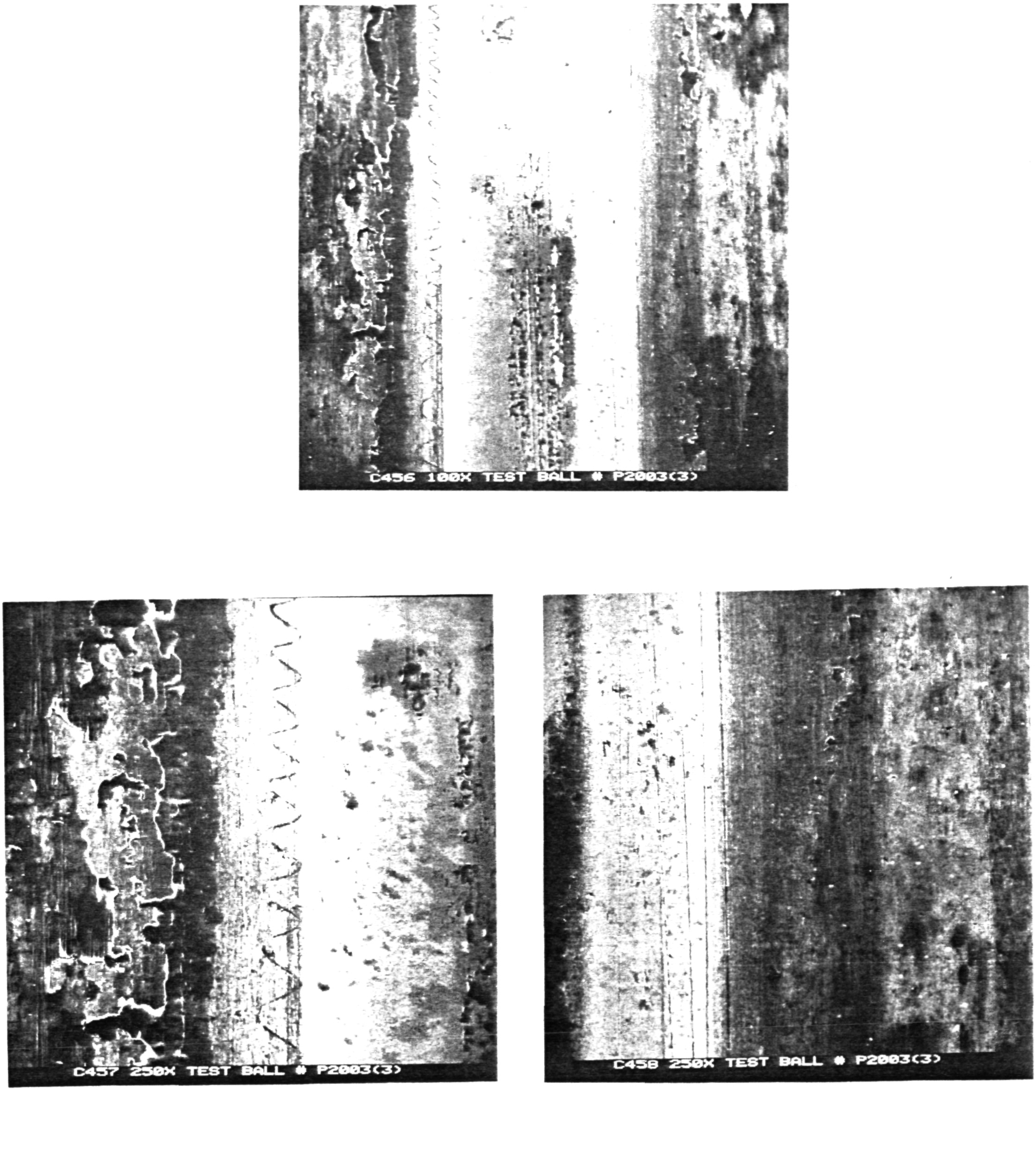

FIGURE 24: SEM PICTURES OF BALL WEAR TRACK FOR THE TEST IN FIGURE 23

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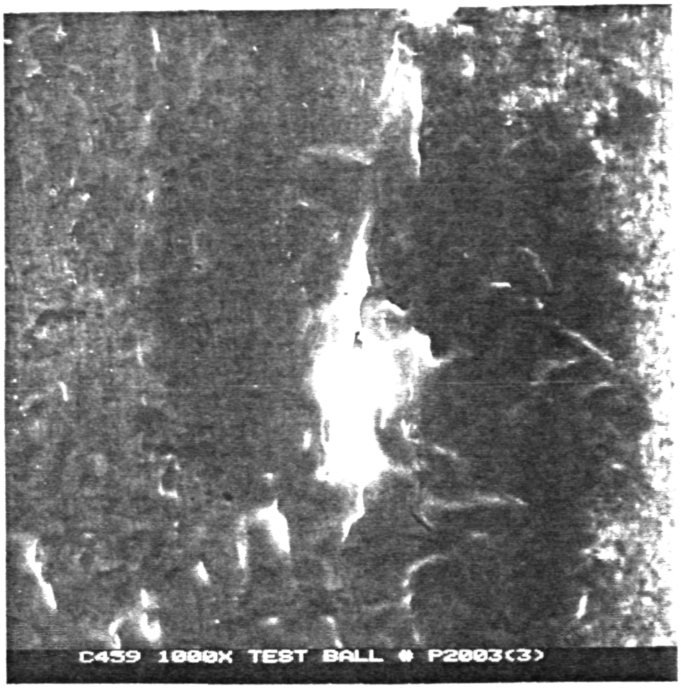


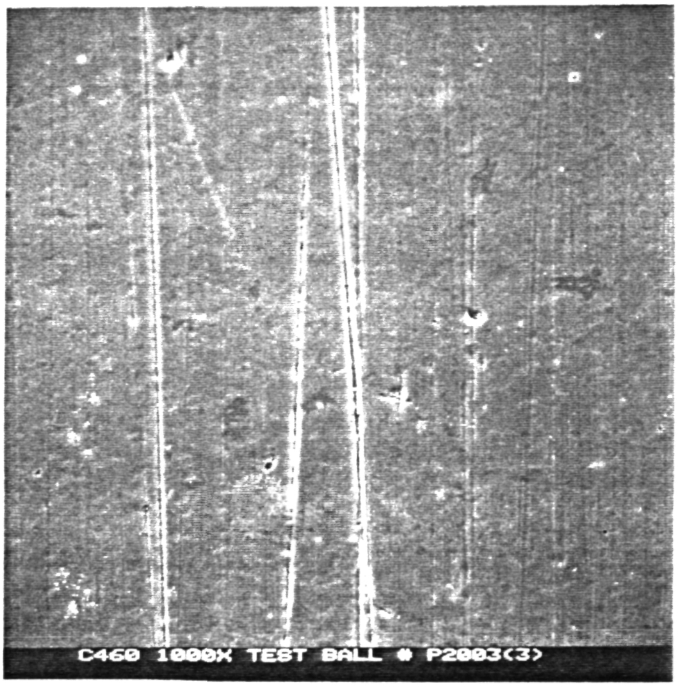
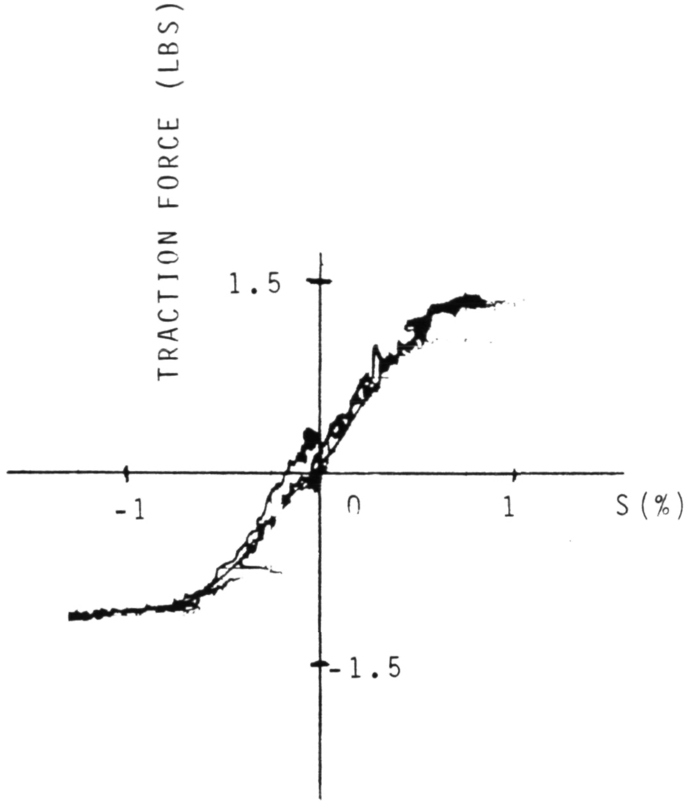


FIGURE 24: SEM PICTURES OF BALL WEAR TRACK FOR THE TEST IN FIGURE 23 (CONTINUED)



LUBRICANT: P03AG
NORMAL LOAD: 12 LBS
CONTACT STRESS: 150 KSI
ROLLING SPEED: 300 IPS

FIGURE 25: TRACTION CURVE FOR P03AG GRAPHITE AT 300 IPS ROLLING SPEED AND 150 KSI CONTACT STRESS

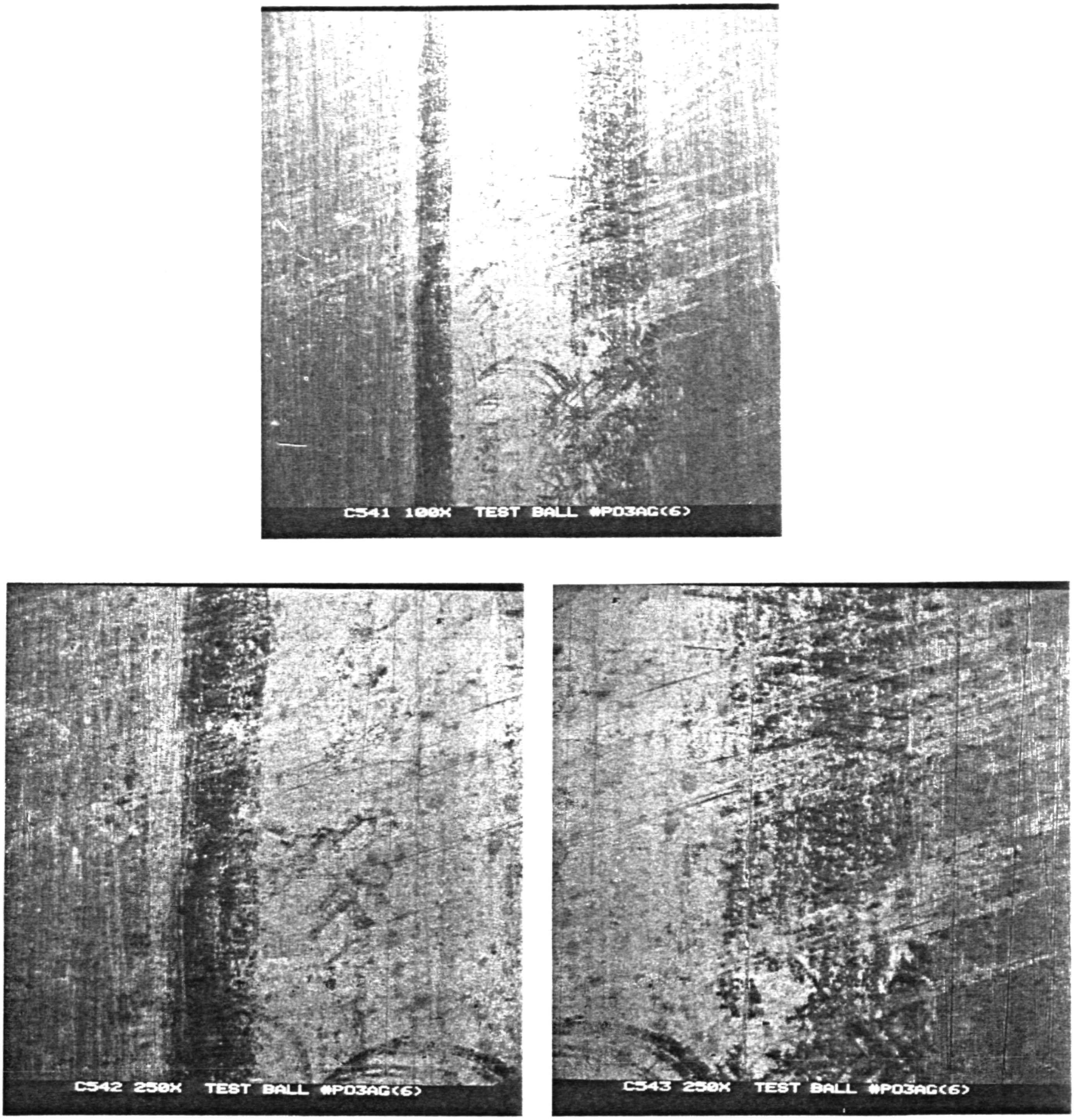
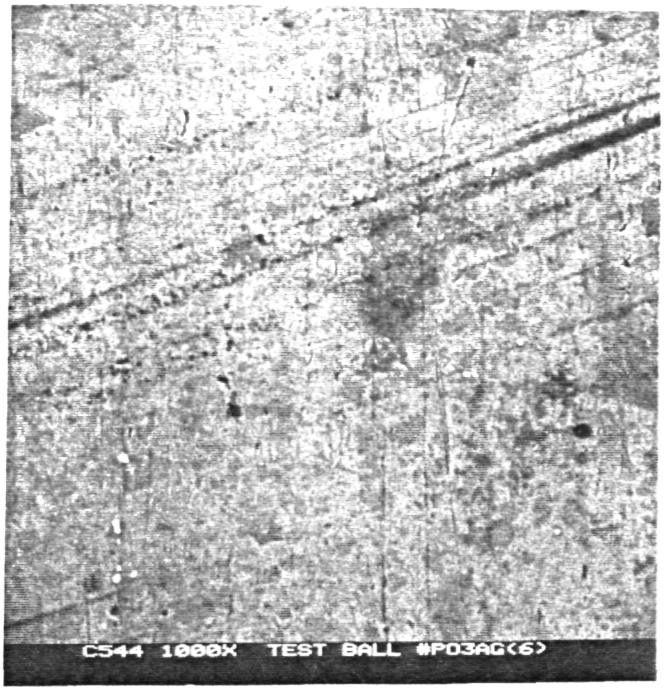


FIGURE 26: SEM PICTURES OF BALL WEAR TRACK FOR THE TEST IN FIGURE 25



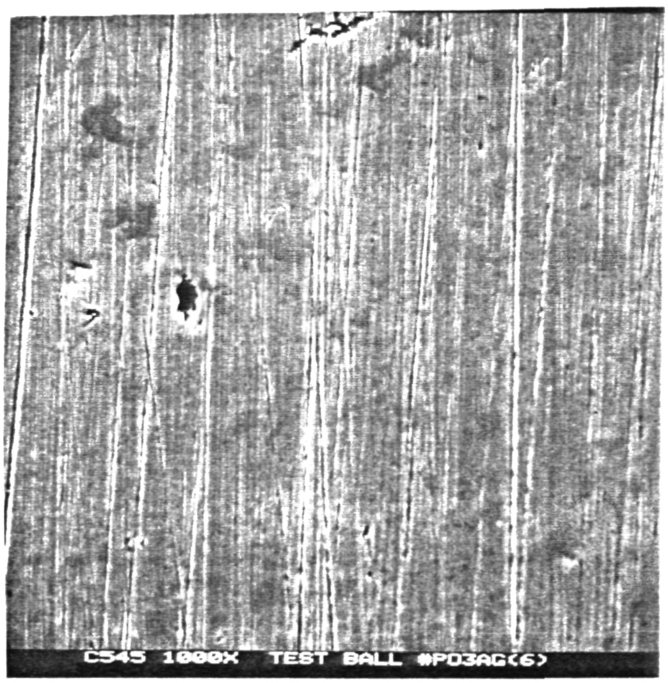


FIGURE 26: SEM PICTURES OF BALL WEAR TRACK FOR THE TEST IN FIGURE 25 (CONTINUED)

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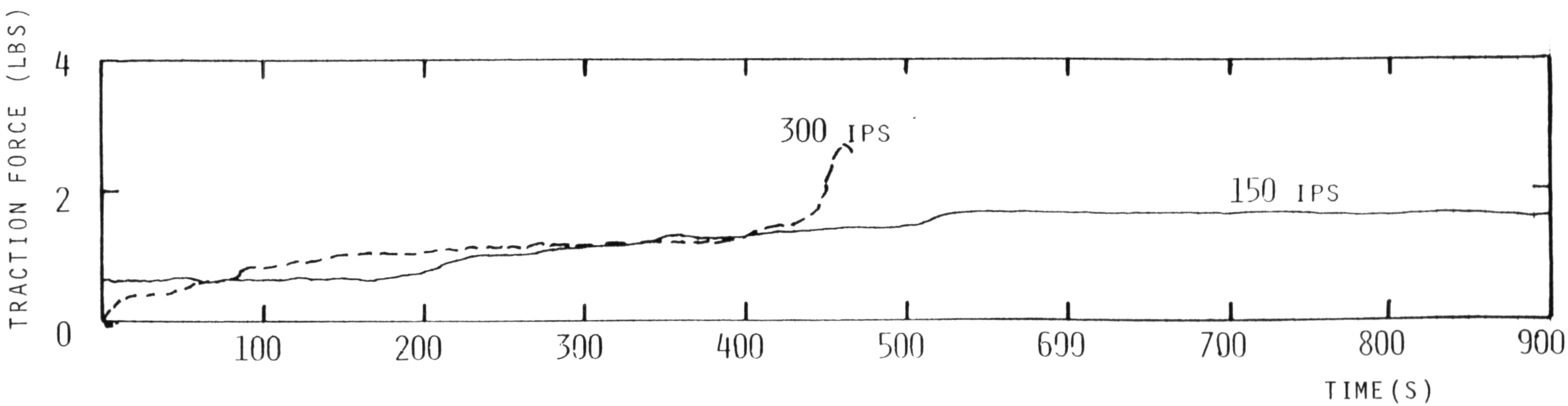
tion curves are shown in Figures 22, 24 and 26. The figures illustrate the differences in the way the lubricants transfer from the disc to the ball. P2003 was the best at protecting the contact surfaces at 300 ksi contact stress. Some surface damage was evident for all the graphites at 300 ksi contact stress and 300 ips rolling speed. The limiting traction force was reached at 1 to 2 percent slide/roll ratio for all the graphites. Extended operation at slide/roll ratio greater than the value at which the limiting traction is reached resulted in wear damage of the contact surfaces.

Traction vs. time curves for P03Ag at 150 ksi contact stress, 0.7% slide/roll ratio and rolling speeds of 150 and 300 ips are shown in Figure 27. The test at 300 ips was stopped when the traction force level rose rapidly. Examination of the ball surface after the test showed extensive wear in the ball track. The ball track for the test at 150 ips did not show significant wear after 15 minutes.

Tests using a fused silica disc were attempted in order to observe the transfer of lubricant from the lubricant specimen to the disc and the flow of lubricant through the contact. Graphite lubricants were used in these tests. The attempts had to be abandoned because the graphites did not burnish well on the



LUBRICANT: P03AG
NORMAL LOAD: 12 LBS
CONTACT STRESS: 150 KSI
ROLLING SPEED: 150 AND 300 IPS
SLIDE/ROLL RATIO: 0.7 %

FIGURE 27: TRACTION FORCE VS. TIME CURVES FOR P03AG GRAPHITE AT 150 KSI CONTACT STRESS, SLIDE/ROLL RATIO OF 0.7% AND ROLLING SPEEDS OF 150 AND 300 IPS.

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smooth, glazed surface of the fused silica disc. Steel disc specimens were examined using an optical microscope after burnishing with P9 graphite. The burnished lubricant was pressed into the valleys on the surface, forming a tough layer which could not be removed by washing with acetone or scraping. Tops of a few asperities protruded above the layer of lubricant. This layer of lubricant wears in preference to the ball and disc surfaces in contact. As long as the lubricant supplied by burnishing the disc surface is sufficient to replace the layer worn away in the contact, the surfaces are protected from wear. Tests with graphite lubricants showed that lubricant replenishment by burnishing was adequate for low slide/roll ratios. The slide/roll ratio at which wear starts appeared to be a strong function of contact stress and a weak function of rolling speed. The threshold value of slide/roll ratio decreased as both load and rolling speed increased. There was insufficient data to derive a quantitative relationship between the threshold slide/roll ratio and the operating conditions.

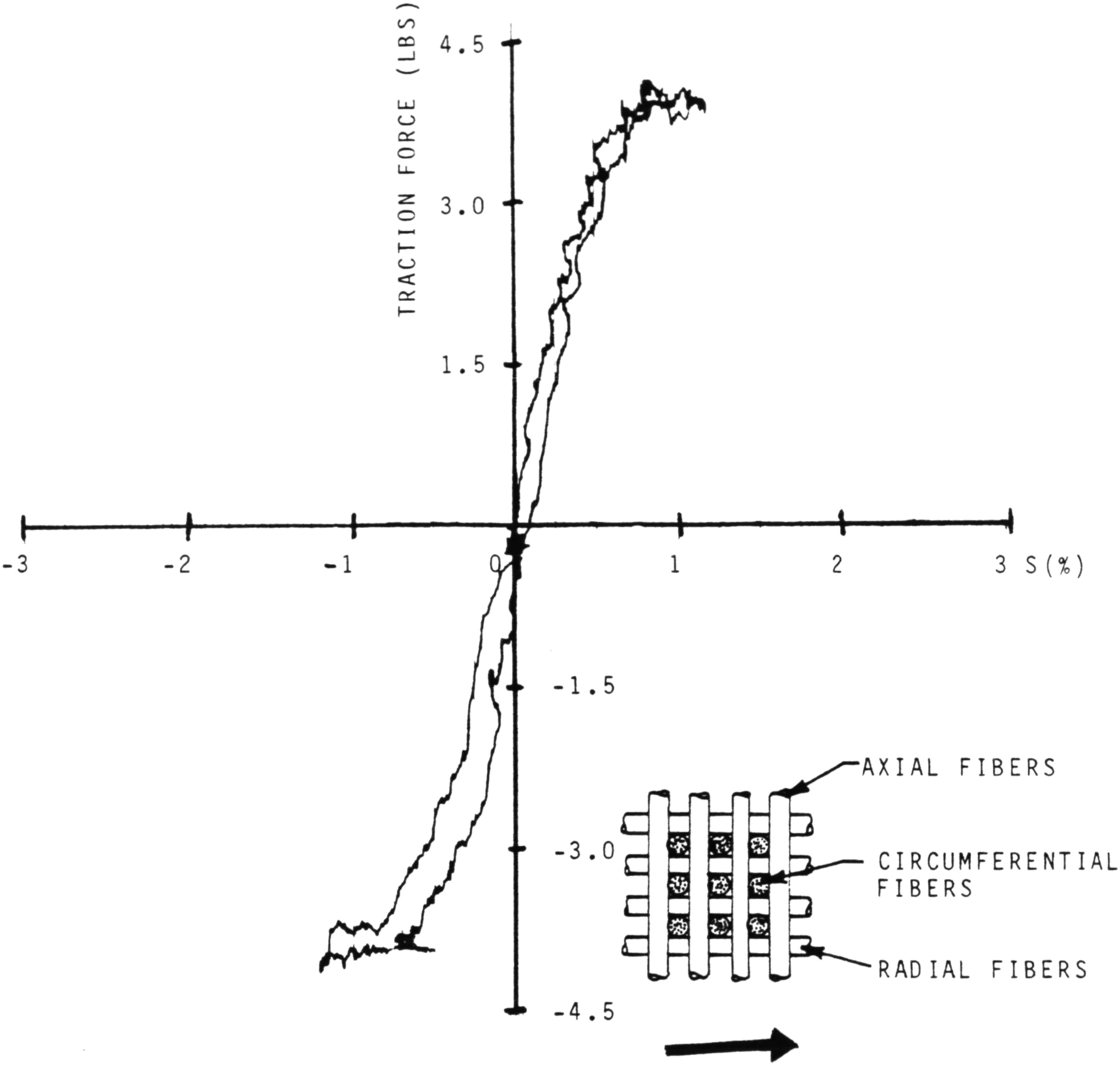
4.30 Composite Lubricants

All five composite lubricants tested at room temperature were polyimide based lubricants and are, therefore, limited to applications below 600°F. As with graphite lubricants, the 250 ksi

contact stress value in the test matrix was not used for screening tests.

Both Polybon L and Polybon M proved to be poor lubricants under the operating conditions used in the tests. While Polybon M protected the contact surfaces better than Polybon L, both of them allowed unacceptable levels of wear even at 150 ksi contact stress. Discussions with Tribon Company's engineers indicated that the two materials were not meant for use under the conditions used in the test program.

Both HAC1/T50F3 and HAC1/T50F4, filled with Westinghouse Compact, performed well at 150 ksi contact load. At 300 ksi contact stress, HAC1/T50F3 provided marginal wear protection at 150 ips rolling speed and failed completely at 300 ips. HAC1/T50F4 provided good wear protection at 150 ips rolling speed. Because of time constraints, it was not tested at 300 ips rolling speed. Traction curve for HAC2A/T50F4 for a 150 ksi contact stress and 300 ips rolling is shown in Figure 28 with the SEM pictures of the ball specimen in Figure 29. The lubricant appears to have formed a thin, uniform layer which adheres to the ball surface. The behavior appears to be similar to that of polyimide films above the transition temperature required for good wear protection [30,31]. The HAC2A/T50F4 composite, filled with Turbolube,



LUBRICANT: HAC2A/T50F4
NORMAL LOAD: 12 LBS
CONTACT STRESS: 150 KSI
ROLLING SPEED: 300 IPS

DIRECTION OF SLIDING OF DISC SURFACE RELATIVE TO THE 3-D WEAVE

FIGURE 28: TRACTION CURVE FOR HAD2A/T50F4 COMPOSITE AT 150 KSI CONTACT STRESS AND 300 IPS ROLLING SPEED. THE INSERT IS VIEW Y FROM FIGURE 12.

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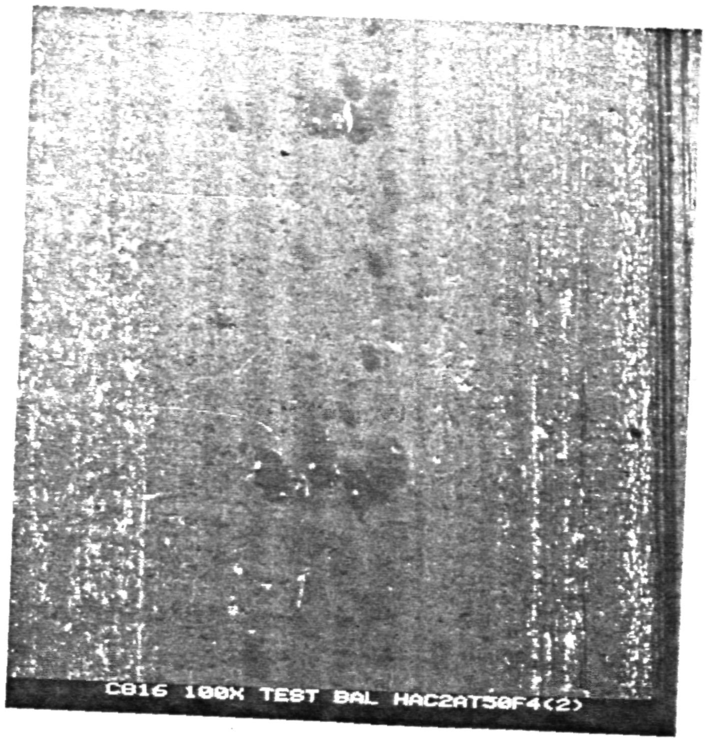

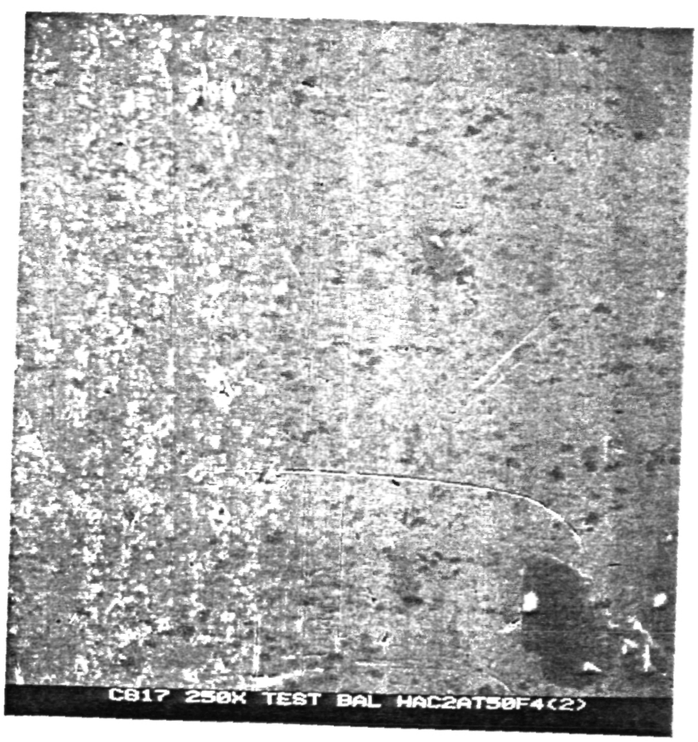

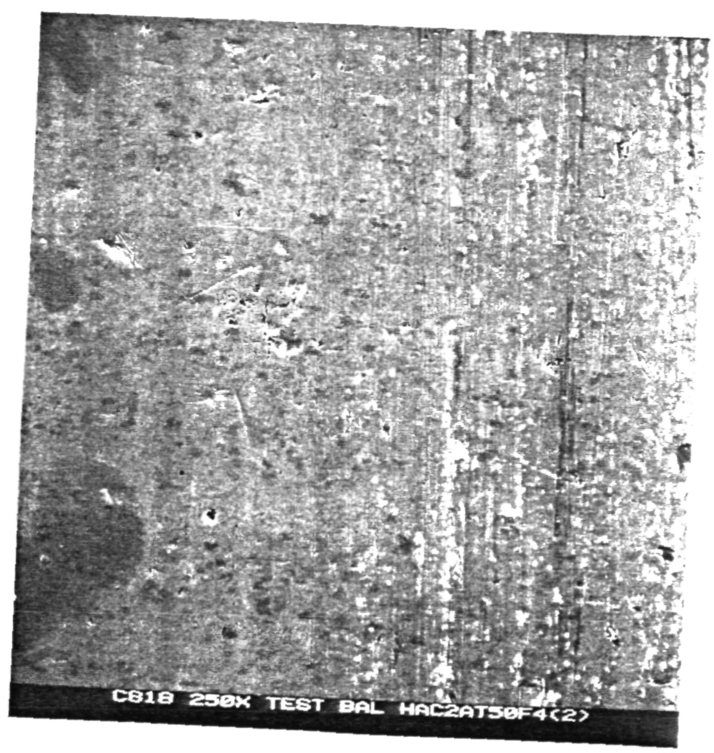

FIGURE 29: SEM PICTURES OF BALL WEAR TRACK FOR TEST IN FIGURE 28

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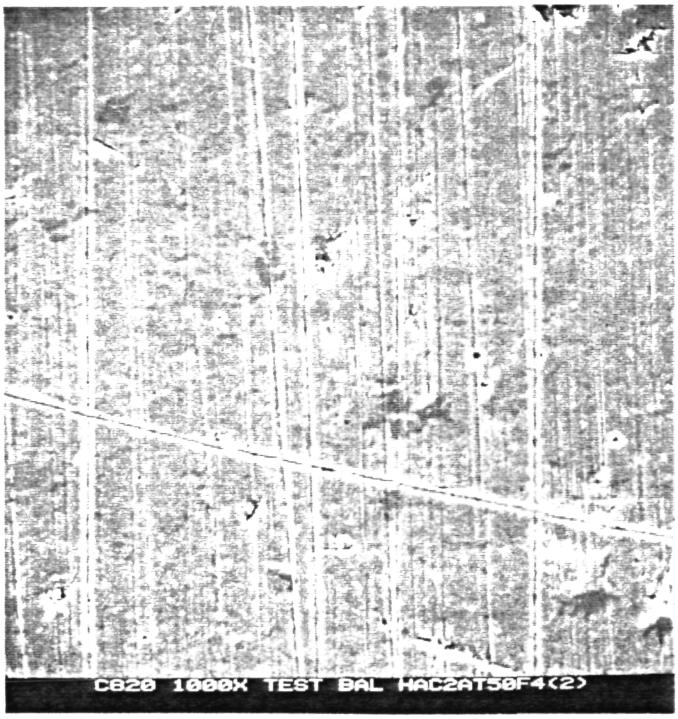

FIGURE 29: SEM PICTURES OF BALL WEAR TRACK FOR TEST IN FIGURE 28 (CONTINUED)

F-4185C-1-R100

provided good wear protection at all load and speed combinations tested except 300 ksi contact stress and 300 ips rolling speed. The appearance of the lubricant film on ball specimens for the successful tests with HAC1 lubricants was the same as shown in Figure 29. The values of limiting traction force for HAC1 and HAC2 composites are listed in Table 8.

4.40 Summary

Based on the screening tests described in this section, four lubricants were selected for further testing using the complete test matrix: P03Ag, P3310 and P2003 graphites and HAC2A/ T50F4 composite. The results are described in the next section.

TABLE 8: LIMITING TRACTION FORCE FOR HUGHES AIRCRAFT COMPANY COMPOSITES

LUBRICANT	ROLLING SPEED (IPS)	NORMAL LOAD (LB)	MAXIMUM HERTZ CONTACT STRESS (KSI)	LIMITING TRACTION FORCE (LB)
HAC1/T50F3	150	12	150	0.7
	300	12	150	1.5
	150	93	300	19.7
	300	93	300	21.8
HAC1/T40F4	150	12	150	2.3
	300	12	150	3.4
	150	93	300	21.9
	300	93	300	----
HAC2A/T50F4	150	12	150	4.2
	300	12	150	3.9
	150	93	300	19.7
	300	93	300	24.6

5.0 TEST RESULTS FOR SELECTED LUBRICANTS

5.10 Test Results

The rolling element-lubricant combinations tested using the full test matrix are listed in Table 9. A traction curve was obtained using the standard test procedure and then, using the same ball specimen, a traction vs. time plot was generated at a slide/roll ratio that would yield a starting traction force equal to half the limiting traction force given by the traction curve. This procedure was adopted to save time since each test using a fresh track on disc and a new ball specimen required an average of 4 to 5 hours. The ball specimens were not examined in the SEM for tests with M50 discs since the ball surface during screening tests had always shown wear after traction vs. time curves in the presence of sliding. Ball specimens from tests with silicon nitride discs were examined in the SEM because these were the first tests run with silicon nitride discs and no previous experience was available to surmise the condition of the ball surfaces.

A short trial program was undertaken with silicon nitride and zirconia discs to verify that the test procedure developed for steel discs would also work with ceramic dics and, if necessary, to develop a suitable test procedure for ceramics. The results

TABLE 9: ROLLING ELEMENT/LUBRICANT COMBINATIONS TESTED

BALL	DISC	LUBRICANTS
M50	M50	PO3Ag, P3310, P2003, HAC2A/T50F4
M50	NC132	P2003, HAC2A/T50F4
M50	PSZ	HAC2A/T50F4

showed that the standard test procedure developed for steel discs would work for silicon nitride. However, all attempts to burnish lubricant on the zirconia discs failed. Only HAC2A/T50F4 provided sufficient lubricant transfer to attempt obtaining traction curves using zirconia discs. The values of limiting traction force for tests at a rolling speed of 300 ips are plotted in Figure 30. An examination of the zirconia disc surface showed extensive wear at all loads. Further testing with zirconia discs was abandoned.

Values of limiting traction force for tests using M50 discs are plotted in Figures 31, 32, 33 and 34 for P03Ag graphite, P3310 graphite, P2003 graphite and HAC2A/T50F4 composite, respectively. The results show that the limiting traction force was a function of the contact stress and rolling speed; the effect of contact stress was greater than that of rolling speed. The traction forces for all the lubricants were about the same magnitude. The acceptability of a lubricant should, therefore, be based on its ability to limit wear of surfaces in contact. HAC2A/T50F4 provided the best wear protection, followed by P2003, P3310 and P03Ag. These lubricants are, however, designed for use at high temperatures and the behavior of the lubricants at room temperature may not be indicative of their capabilities at high temperatures. The shape of the traction curves was the same

BALL MATERIAL: M50
DISC LUBRICANT: ZIRCONIA (PSZ MS GRADE)
LUBRICANT: HAC2A/T50F4

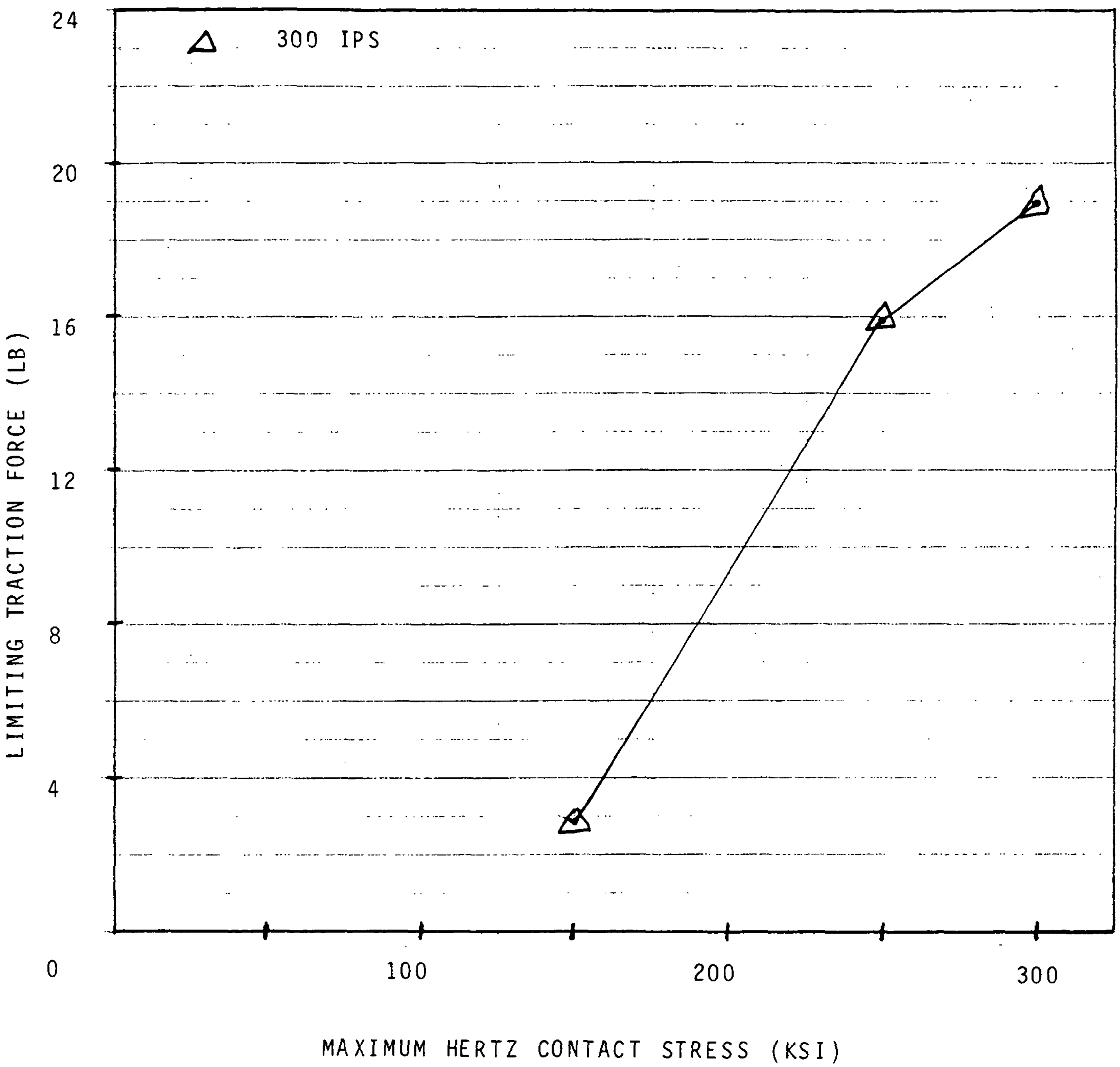

FIGURE 30: LIMITING TRACTION FORCES FOR HAC2A ON ZIRCONIA DISC

BALL MATERIAL: M50
DISC LUBRICANT: M50
LUBRICANT: P03AG GRAPHITE

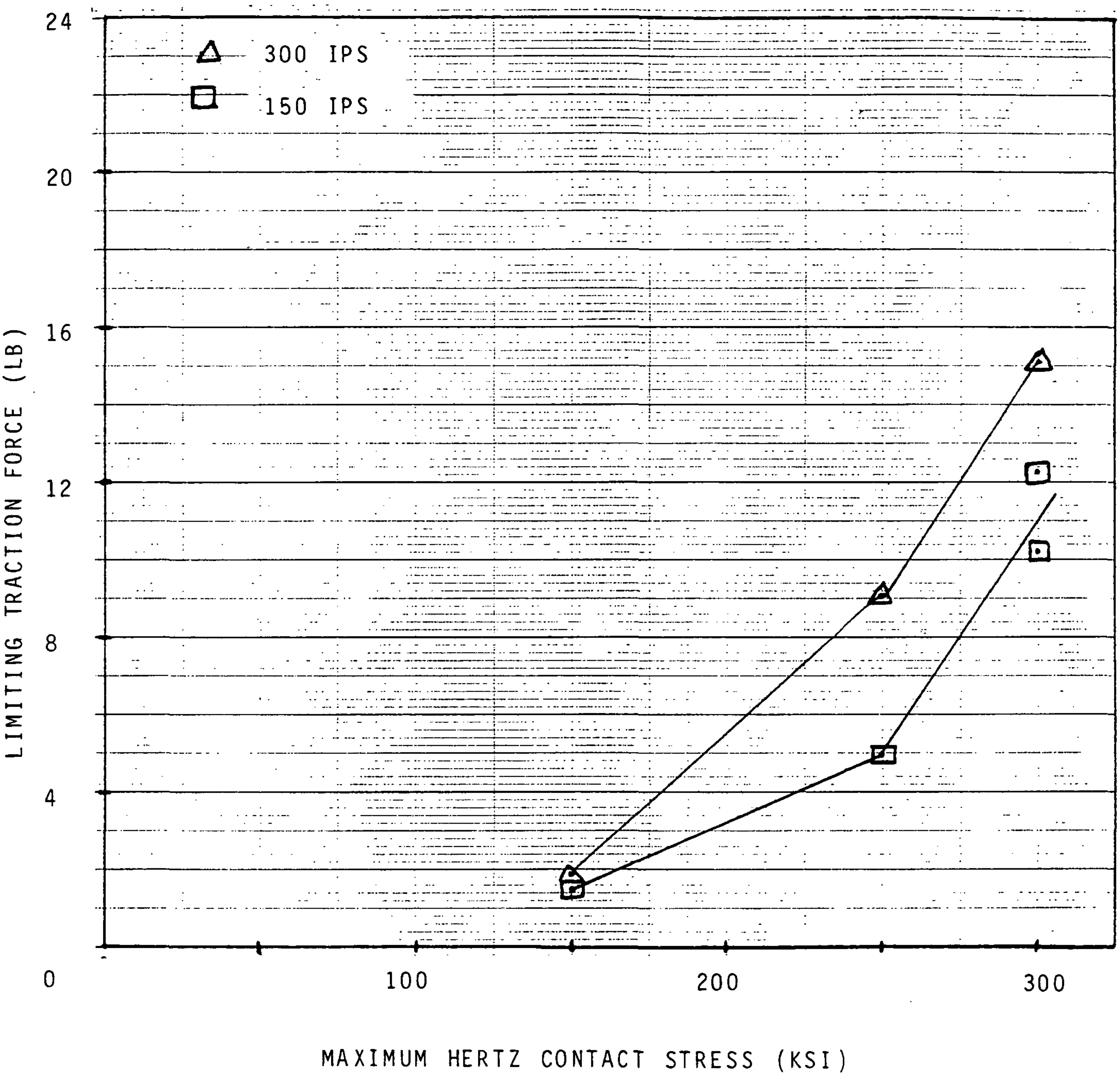

FIGURE 31: LIMITING TRACTION FORCES FOR P03AG ON M50 STEEL DISC

BALL MATERIAL: M50
DISC LUBRICANT: M50
LUBRICANT: P3310 GRAPHITE

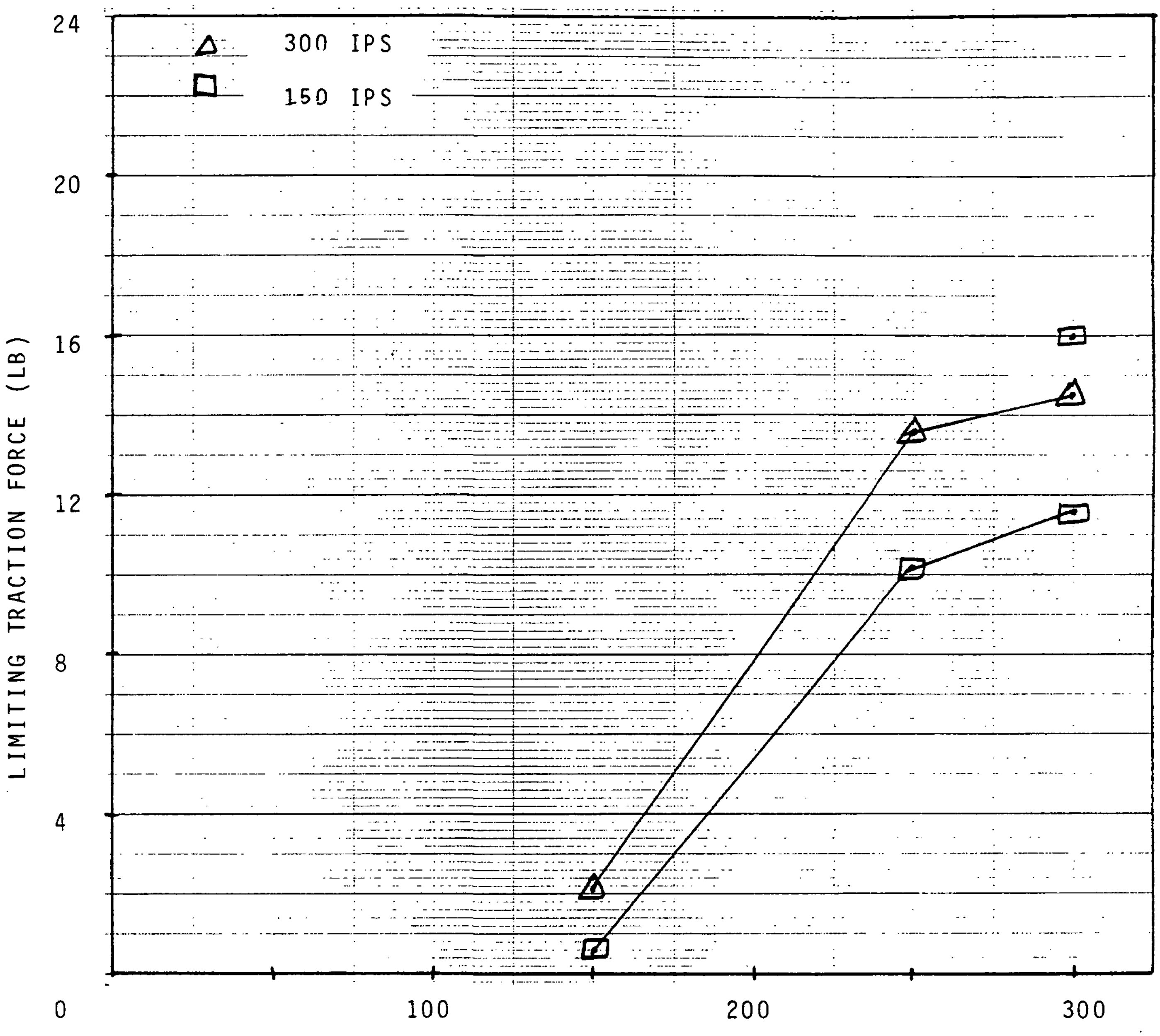

FIGURE 32: LIMITING TRACTION FORCES FOR P3310 ON M50 STEEL DISC

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BALL MATERIAL: M50
DISC LUBRICANT: M50
LUBRICANT: P2003 GRAPHITE

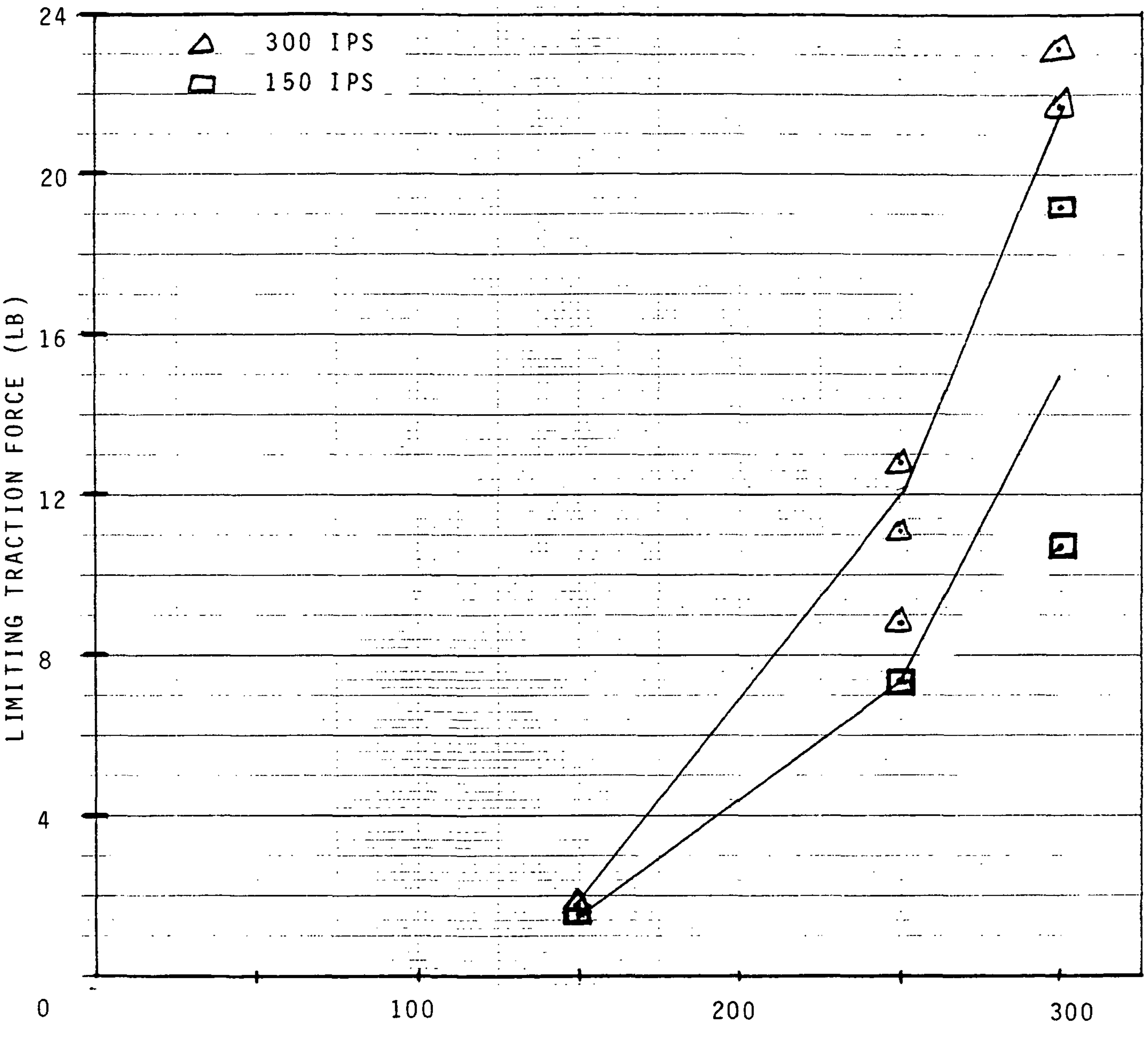


FIGURE 33: LIMITING TRACTION FORCES FOR P2003 ON M50 STEEL DISC

BALL MATERIAL: M50
DISC LUBRICANT: M50
LUBRICANT: HAC2A/T50F4

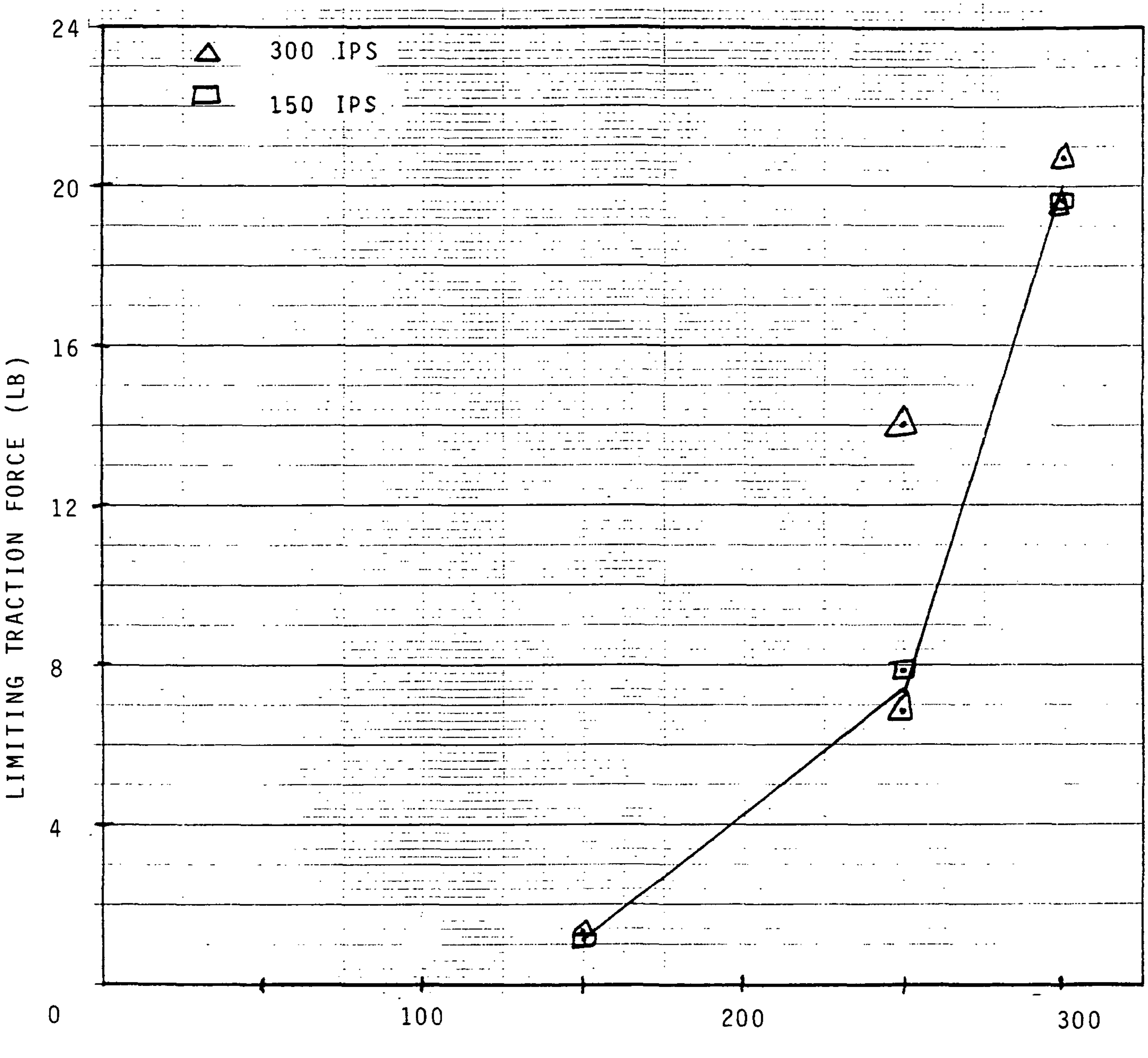


FIGURE 34: LIMITING TRACTION FORCES FOR HAC2A ON M50 STEEL DISC

as those obtained during screening tests (see Section 4.0), with the limiting traction force being reached at a slide/roll ratio of 1 to 2 percent. The magnitudes of the traction forces were also comparable to those observed during screening tests, except for tests at 150 ksi contact stress using P2003 and HAC2A/T50F4 lubricants which yielded traction forces lower than those during screening tests.

Values of limiting traction force for tests using silicon nitride discs are plotted in Figures 35 and 36, respectively, for P2003 and HAC2A/T50F4 composite lubricants. The trends were the same as those observed with M50 discs but the traction forces were lower than those obtained with M50 discs. Examination of disc surfaces using an optical microscope showed good transfer of lubricant to the disc surface and no significant wear at any of the contact stress-rolling speed combination even after 15 minutes of traction vs. time testing with sliding at the contact. SEM pictures of ball specimens from tests with HAC2A/T50F4 showed that the ball surface was protected while traction curves were being generated for all the load-speed combinations but was damaged during the traction vs. time tests at 250 ksi-300 ips and 300 ksi-150 ips combinations. No traction vs. time test was run at the 300 ksi-300 ips combination because of surface wear at the lower load-speed combinations. The M50 ball-silicon nitride

BALL MATERIAL: M50
DISC LUBRICANT: SILICON NITRIDE (NC132)
LUBRICANT: P2003 GRAPHITE

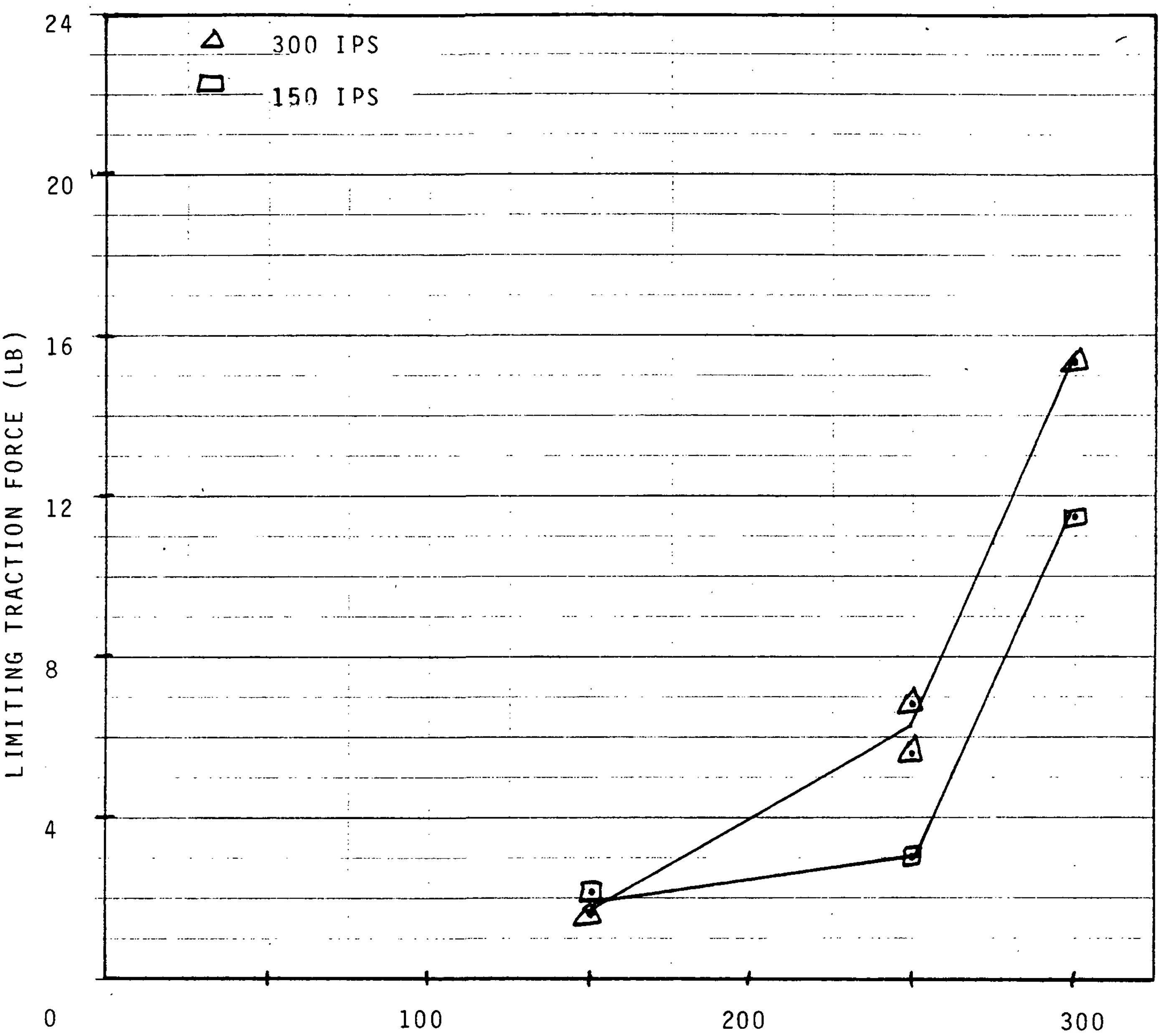


FIGURE 35: LIMITING TRACTION FORCES FOR P2003 ON SILICON NITRIDE DISC

BALL MATERIAL: M50
DISC LUBRICANT: SILICON NITRIDE (NC132)
LUBRICANT: HAC2A/T50F4

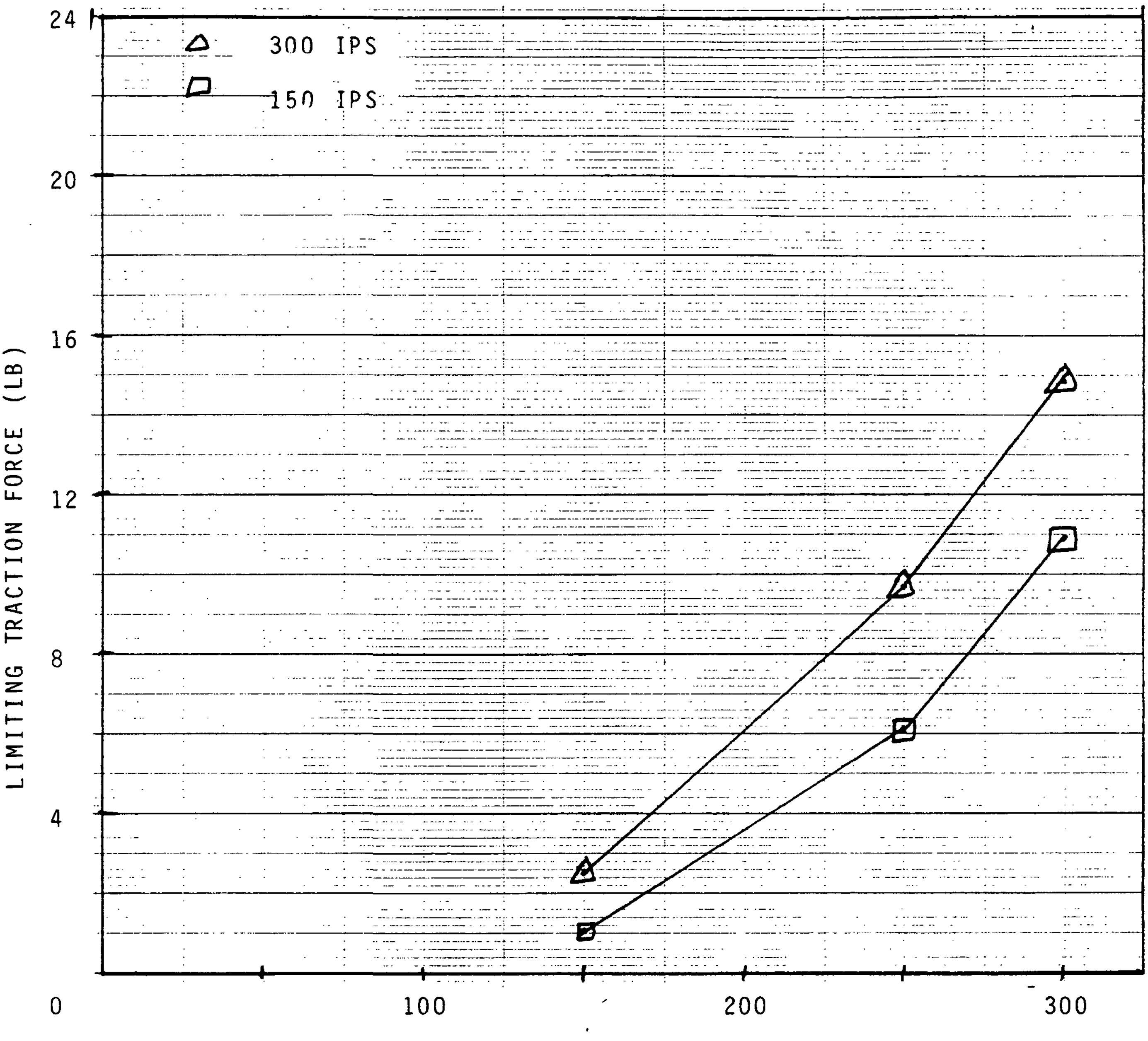


FIGURE 36: LIMITING TRACTION FORCES FOR HAC2A ON SILICON NITRIDE DISC

disc material combination yielded less wear at the contact surfaces than the M50 ball-M50 disc and 440C ball-M50 disc combinations. A number of solid lubricated bearings used in high speed turbine engines have been hybrid bearings using silicon nitride balls with steel races. The choice of a hybrid bearing over an all-steel bearing is in agreement with the results of the current test program because of lower traction forces and better wear performance for hybrid contacts over all-steel contacts. The results suggest that an all-ceramic contact may provide good wear performance over the operating conditions anticipated in advanced adiabatic diesel engines. The possibility will be examined in the next phase of the program by testing silicon nitride balls against silicon nitride discs.

5.20 Summary

The test program showed that while the shape of traction curves for solid lubricated contacts is similar to those for liquid lubricants, the magnitude of traction forces is at least an order of magnitude higher than traction forces for liquid lubricants. A solid lubricated contact will, therefore, generate more heat at the contact. Dissipation of the heat generated in a solid lubricated bearing is a major problem that must be dealt with at the design stage. Solid lubricated bearings are not low fric-

tion bearings when compared with oil lubricated bearings. They should, therefore, be used only when environmental factors preclude the use of liquid lubricants.

6.0 ANALYTICAL MODELS AND GENERAL COMMENTS

6.10 Analytical Models

The purpose of analytical models is to enable prediction of traction forces at a solid lubricated contact for the lubricants tested in the test program. The form for the equations was selected to allow simulation of solid lubricated contacts using the computer program TRIBOS [42] developed at SKF to assess the performance of a concentrated rolling/sliding contact. The traction curve parameters required as input are the slope m of the traction curve at low slide/roll ratios and the limiting traction force T_L at high slide/roll ratios (see Figure 37). The values of the limiting traction force and the slope are listed in Table 10. The form of equations generated using the data is given by

$$T_L = c_1 U^{a_1} p^{b_1} \quad (3)$$

$$m = c_2 U^{a_2} p^{b_2} \quad (4)$$

where U is the rolling speed, p the maximum Hertz contact stress and a_1, b_1, c_1, a_2, b_2 and c_2 are constants obtained by curve fitting the data in Table 10. The values of the constants are listed in Table 11. The statistical measures generated by the curve fitting software confirmed an earlier observation that the

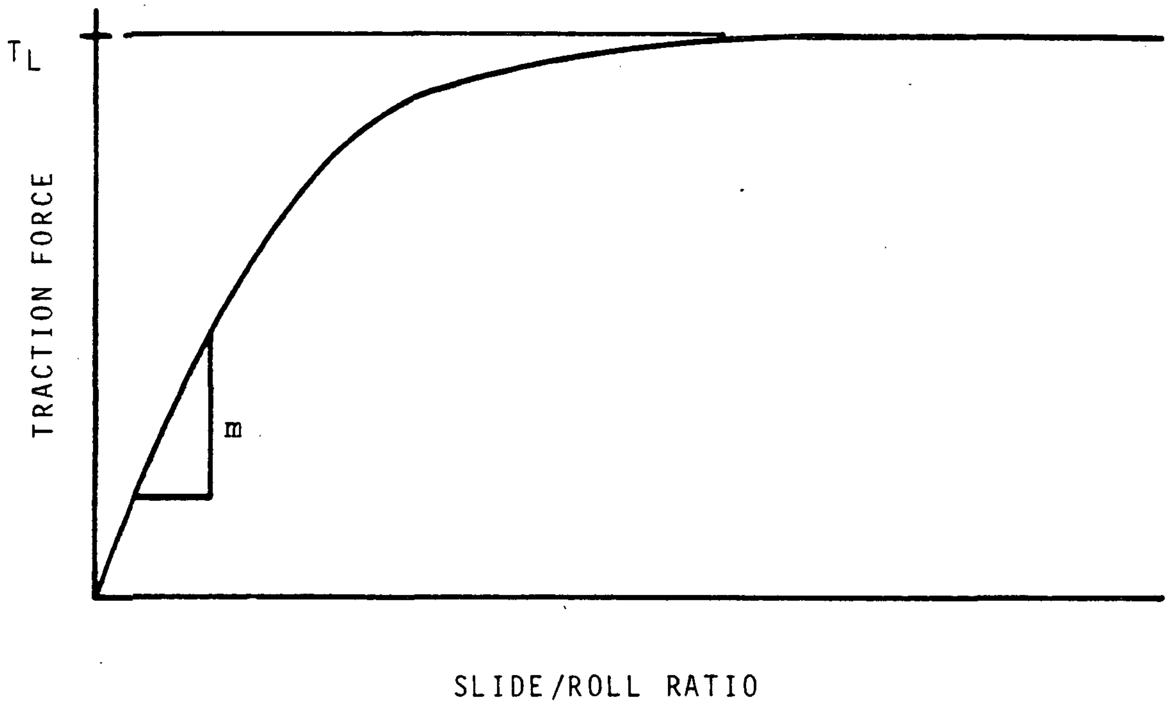


FIGURE 37: DEFINITION OF TERMS FOR ANALYTICAL MODEL

TABLE 10: DATA USED FOR ANALYTICAL MODEL

LUBRICANT	ROLLING SPEED (ips)	NORMAL LOAD (lb)	MAXIMUM HERTZ CONTACT STRESS (ksi)	LIMITING TRACTION FORCE (lb)	SLOPE m (lb)
P03AG (M50 Disc)	150	12	150	1.5	195
	300	12	150	1.9	667
	150	53	250	5.0	641
	300	53	250	9.1	1573
	150	93	300	10.2	1644
	150	93	300	12.2	2041
	300	93	300	15.1	2662
P3310 (M50 Disc)	150	12	150	0.6	102
	300	12	150	2.1	412
	150	53	250	10.2	2161
	300	53	250	13.6	2178
	150	93	300	11.6	2268
	150	93	300	16.0	3207
	300	93	300	14.5	2541
P2003 (M50 Disc)	150	12	150	1.6	213
	300	12	150	1.7	233
	150	53	250	7.4	1197
	300	53	250	12.8	1624
	300	53	250	8.8	1020
	300	53	250	11.1	1282
	150	93	300	10.5	3313
	150	93	300	19.2	2707
	300	93	300	21.8	1450
	300	93	300	23.2	2736

NOTE: M50 BALLS USED FOR ALL TESTS

TABLE 10: DATA USED FOR ANALYTICAL MODEL (CONTINUED)

LUBRICANT	ROLLING SPEED (ips)	NORMAL LOAD (lb)	MAXIMUM HERTZ CONTACT STRESS (ksi)	LIMITING TRACTION FORCE (lb)	SLOPE m (lb)
HAC2A/ T50F4 (Si_3N_4 Disc)	150	8	150	1.1	116
	300	8	150	2.6	480
	150	36	250	6.1	1281
	300	36	250	9.7	1539
	150	63	300	10.9	1885
	300	63	300	14.9	2235
P2003 (Si_3N_4 Disc)	150	8	150	2.1	426
	300	8	150	1.6	365
	150	36	250	3.0	424
	300	36	250	6.8	1164
	300	36	250	5.7	1247
	150	63	300	11.5	2360
	300	63	300	15.4	2223
HAC2A/ T50F4 (M50 Disc)	150	12	150	1.2	185
	300	12	150	1.3	365
	150	54	250	7.9	1452
	300	54	250	6.8	1644
	300	54	250	14.0	1982
	150	93	300	19.7	3630
	300	93	300	20.7	3575
	300	93	300	19.6	3045

NOTE: M50 BALLS USED FOR ALL TESTS

contact stress had a stronger effect on the limiting traction force than the rolling speed. Multiple R^2 values listed in Table 11 are a measure of the goodness of regression fits. A value of 1.0 implies a perfect fit. The values show a good fit for all the equations.

6.20 General Comments

Life of solid lubricated bearings has been limited by wear at rolling element/bearing race surfaces, loss of bearing clearances due to heat generated in the bearing or a buildup of solid lubricant film and cage failure in high speed bearings. The bearing geometry must be selected to promote formation of a durable lubricant film and minimize heat generation at the contacts. The bearing and housing design should maximize the flow of heat away from the bearing. Some of the design criteria are:

(1) Reduce Contact Stress - This can be achieved by using a larger number of balls or larger sized balls. Lower contact stress will enhance the maintenance of a lubricant film and reduce heat generation by lowering the traction forces at the contact.

(2) Reduce Sliding at the Contact - Heathcote slip at the contact can be reduced by reducing the conformity of

TABLE 11: CONSTANTS FOR ANALYTICAL EQUATIONS

LUBRICANT	DISC MATERIAL	a_1	b_1	c_1	a_2	b_2	c_2	MULTIPLE R^2 FOR a_1 b_1 c_1	MULTIPLE R^2 FOR a_2 b_2 c_2
P03AG	M50	0.518	2.927	4.40×10^{-8}	1.083	2.623	2.0×10^{-6}	0.985	0.915
P3310	M50	0.767	3.858	8.04×10^{-11}	0.636	3.892	2.5×10^{-8}	0.921	0.896
P2003	M50	0.474	3.428	4.45×10^{-9}	-0.197	3.455	1.94×10^{-5}	0.961	0.944
HAC2A/ T50F4	M50	0.150	3.957	1.35×10^{-9}	0.329	3.645	5.16×10^{-7}	0.971	0.976
P2003	Si_3N_4	0.313	2.583	7.36×10^{-7}	0.381	2.233	6.32×10^{-4}	0.833	0.726
HAC2A/ T50F4	Si_3N_4	0.787	2.925	1.08×10^{-8}	0.853	3.201	2.73×10^{-7}	0.984	0.912

NOTE: M50 BALLS USED FOR ALL TESTS

rolling element/raceway contacts. Lubricant film is depleted rapidly by sliding at the contact. Reducing conformity will, however, increase the contact stress. The selection of bearing geometry and materials must strike a balance between criteria (1) and (2).

(3) Wear Resistant Bearing Materials - Use of silicon nitride discs reduced the wear of contact surfaces. Wear resistance can also be provided by coatings such as Armoloy and Titanium Nitride. The coatings can also be used to promote the formation and retention of solid lubricant films.

(4) Heat Dissipation - The bearing and housing design must provide means of dissipating the heat generated at the contacts.

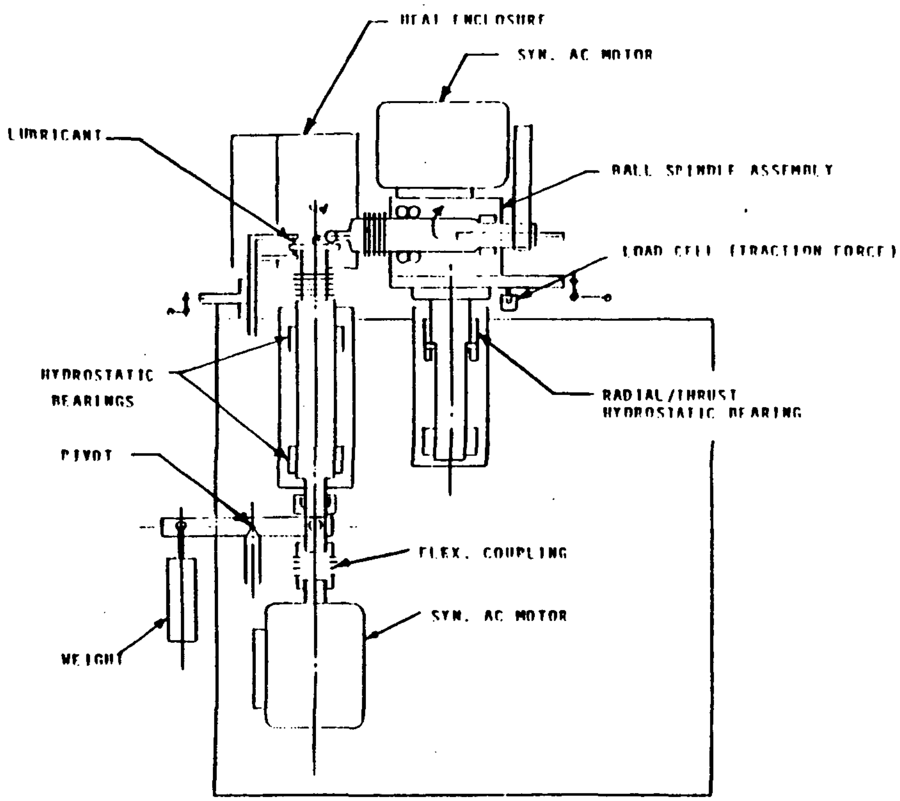
7.0 WORK PLANNED FOR PHASE II

7.10 High Temperature Test Rig

In view of the problems encountered with the modified EHD rig used for room temperature tests, SKF decided to design a new test rig optimized for solid lubricated contacts. A schematic of the test rig is shown in Figure 38. The rig is now being manufactured.

7.20 High Temperature Tests

The four lubricants selected in the room temperature tests - P03AG, P3310, P2003 graphites and HAC2A/T50F4 composite - will be tested at high temperature. The analytical models described in Section 6.10 will be refined to incorporate temperature effects. Silicon nitride balls will be used for most of the tests. Smaller disc specimens are being manufactured to allow SEM examination of disc surfaces. Use of Armoloy coating on M50 ball and disc specimens is being considered as a possible means of improving lubricant transfer and reducing wear.



TEMPERATURE	RT - 1200°F 1500°F POTENTIAL
CONTACT STRESS	0 - 300 KSI
TRACTION	0 - 100 LBS
ROLLING SPEED	0 - 1500 FT/MIN
NORMAL LOAD	0 - 100 LBS
SLIDE/ROLL RATIO	0 - 10%

FIGURE 38: HIGH TEMPERATURE TEST RIG

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